ECONOMY
IN THE USE OF STEAM.

FRANK SALTER.

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THE USE OF STEAM:

A STATEMENT OF THE PRINCIPLES ON WHICH A SAVING OF STEAM CAN BEST BE EFFECTED.

BY

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PREFACE.

THIS little book deals with the *use* of steam as distinguished from its *production*. It is an attempt to state the principles laid down by theoretical writers,—Clausius, Rankine, and others, who are better known in the class-room than in the office,—in such a form as to be useful to practical Engineers; to test by these principles the modes of working which have been found most advantageous in practice; and to offer here and there some new suggestions. Mathematics have been dispensed with as far as possible; and where they could not be altogether avoided, a smaller type has been used in order that the reader who wishes to pass over the mathematical parts may do so conveniently. The principal conclusions have been printed in italics to attract the attention of any who have not time to read the whole. The earlier chapters refer especially to the *theory* of the working of steam, and the others to the application of the theory. The later chapters will therefore have the most interest for Engineers, but they will be more readily followed if the early ones are not entirely passed over.

ECONOMY IN THE USE OF STEAM.

CHAPTER I.

INTRODUCTION.

BEFORE entering upon the subject of the action of steam in the cylinder of an engine, a few preliminary observations will be necessary to guard against mistakes which sometimes arise with reference to the scales of measurement of pressures, temperatures, &c.

ABSOLUTE PRESSURE.—The usual mode of denoting pressure of steam, namely, in pounds above the atmospheric pressure, is simple and convenient for most purposes relating to the boiler, but leads to misconception when applied to steam in the cylinder of an engine. Water evaporated in the open air is said, according to this notation, to be transformed into steam of zero pressure, instead of steam of 14·7 lbs. pressure per square inch, which pressure counterbalances that of the atmosphere. If such steam is used in a condensing engine, the effect is said to be due to *vacuum*, which is still regarded by some people as a separate force unconnected with the steam, and in fact operating on the other side of the piston. When steam of higher pressure is used, it is customary in finding the horsepower to add the *vacuum* to the *steam pressure*, so

carrying out the same idea. It need hardly be remarked, however, that the employment of the condenser is simply the removal of an opposing force, and therefore the condenser ought to be regarded not as a manufacturer of power, but as a contrivance by means of which all the work performed by the steam in the cylinder is brought to bear on the piston-rod, instead of some of it being thrown away in pressing back the atmospheric air. The pressure or tension of the steam in a non-condensing engine is resisted by two forces, namely, the strain in the piston-rod and the atmospheric pressure on the other side of the piston. In the condensing engine, however, the opposing force is only the strain in the rod *plus* a pound or two of "back pressure" in the condenser due to the imperfection of the vacuum.

It is common in scientific books to give the pressure of a gas in terms of atmospheres, one atmosphere being a pressure of 14·7 lbs. on the square inch. By this system a pressure of 60 lbs. per square inch by the pressure gauge would be called five atmospheres in all. Here the misconception just alluded to is avoided, and also a simple standard of reference is obtained instead of one having two terms, namely, *pounds* per square *inch* or *foot*. In engineering calculations, however, it is necessary to bring the results into these two terms; and it is most convenient therefore to use them throughout, remembering that *the real or absolute pressure, namely, that above the true vacuum, must always be taken instead of that given by the gauge, the former being*

obtained from the latter by adding 14·7 *lbs., or approximately* 15 *lbs.*

Temperature.—Troublesome as the commonly used scale of pressure as given by the gauge is, the Fahrenheit scale of temperature is even worse. Its inconveniences must be submitted to until the Centigrade scale (in which the freezing point is taken as the zero, and the temperature of water boiling under the pressure of one atmosphere as 100°) has come into general use, instead of being confined, as at present, to pure science. At all events, whenever the metric system of weights and measures is adopted in this country, the Centigrade scale will be received as part of it.

Professor Rankine has suggested the hypothesis of an *absolute* or true zero of temperature existing at *minus* 461·2° Fahr.; or 493·2° below the freezing point. The reason for taking this zero is, that, reckoning from it, the temperature of a gas varies as its volume, the pressure being supposed constant. The original form in which this law was stated was, that if the temperature of a gas is altered, the final volume is to the original, as the final temperature *plus* 461·2° Fahr. is to the original temperature *plus* 461·2° Fahr.; and Professor Rankine's hypothesis certainly simplifies the formula considerably. It is important to remember, however, that it is merely a hypothesis, or rather perhaps a convenient fiction; since the law upon which it is founded is only *very nearly* true, that is to say, the zero which would make it true is not precisely the same for all fixed gases, and even for the same gas is

slightly different at different temperatures. This being understood, *the term "absolute" temperature may be used as meaning the temperature Fahrenheit plus* 461·2°, *which may be regarded in a gas as always proportional to the volume, other circumstances remaining the same.* In this notation the freezing point becomes 493·2° and boiling point at atmospheric pressure 673·2° absolute.

LATENT HEAT.—This convenient and mysterious term is often applied to two ways in which heat disappears, which it is most important to distinguish. For this purpose it will be well to follow out in detail a familiar experiment first made by Dr. Mayer of Heilbronn.

Let a cubic foot of air be taken at 32° Fahr. temperature, and at the atmospheric pressure of 14·7 lbs. on the square inch, or 2116·3 lbs. on the square foot; the weight of this amount will be ·080728 lbs. Let it be heated to a temperature of 525·2° Fahr. without being permitted to expand, that is to say, in a closed vessel. The heat absorbed by the air will be as much as is required to raise the temperature of 6·73 lbs. of water one degree Fahrenheit.

Again, let another portion of air be taken, exactly similar in quantity and circumstances to the former, and let it also be heated to a temperature of 525·2° Fahr., but let it be allowed to expand freely during the operation against a constant pressure of 2116·3 lbs. on the square foot. It will be found to have absorbed this time as much heat as will raise the temperature of 9·476 lbs. of water one degree, or 2·746 lbs. more than in the former case. It is evident that in the first

operation all the heat is expended in raising the temperature of the air, since no change of state such as from solid to liquid takes place, nor is any form of force other than heat the result. In the second operation 6·73 units of heat will have gone, as before, to altering the temperature of the air, but the remaining 2·746 units have ceased to exist as *sensible heat*, that is to say, they are not detectable by the thermometer. Heat which disappears in this manner is often called latent heat, but this is a most misleading use of the term, for what has really taken place is a conversion of heat into another form of force, namely, mechanical motion. The air under consideration was supposed to be heated under a constant pressure, from a nominal temperature of 32° to 525·2°, that is to say, substituting *absolute* for nominal temperature, from 493·2°, to 986·4°. Its volume will therefore have increased in the same proportion, namely, from one cubic foot to two, which involves the lifting of the pressure on one square foot through a space of one foot, that is, 2116·3 foot pounds of work. Dividing this by the units of heat which have disappeared, or 2·746, we obtain 770·7 foot pounds as the equivalent of one unit of heat, a number which differs but slightly from the equivalent found experimentally by Dr. Joule, *viz.* 772.

This coincidence of figures obtained by experiment with those arrived at theoretically, shows that the heat which disappears during the expansion of a gas is entirely transformed into work, and is not at all analogous to the *latent heat* properly so called, which is necessary for the conversion of ice into water, in

which case no external work whatever is done, while no effect is produced which the thermometer can detect. It is true that in the case of the expanded gas, as well as in that of the melted ice, the heat may be made to reappear by the reverse process, but this is not because the heat has remained present, although latent, but because work done in expansion is undone again, the energy "conserved" in the lifted weight becoming actual again by the falling of that weight. In the same way energy is rendered potential or conserved by the raising of a stone from the ground, but reappears in "*vis viva*," actual energy, when the stone is allowed to fall, and ultimately takes the form of heat, which is often visible in a spark, as the stone strikes the ground. The disappearance of heat due to the performance of work is therefore an entirely different matter from the absorption of heat by a body passing from the solid to the liquid state; and if the term "*latent*" is applied to the one, it should not also be used for the other. It is clearly most appropriate to the latter case, since in the process of liquefaction a certain amount of heat is really hidden, remaining in the substance as a secret store. When work is done, however, the heat which is absorbed appears in another form, and passes entirely away from the working substance to that worked upon.

Considering now the evaporation of water, the generation of steam, we shall find that what is commonly called the latent heat of steam, is the sum of the proper latent heat which passes into the steam itself, in order that it may exist as steam, and of a certain amount of

heat which becomes transformed into work, driving away the surrounding medium which resists the expansion of the water into steam.

When a pound of water is evaporated at a temperature of 212° Fahr., and under the atmospheric pressure of 2116·3 lbs. on the square foot, a change of volume takes place from ·016 to 26·36 cubic feet, or an augmentation of volume of 26·34 cubic feet. The formation of a pound of steam, therefore, under these conditions, involves the repulse of the air from 26·34 cubic feet of space, and as this repulse takes place against a pressure of 2116·3 lbs. on the square foot, it follows that an amount of work is performed equal in foot pounds to the product of 26·34 into 2116·3, or 55743. Let this number be divided by the number of foot pounds which are equal in value to one unit of heat, that is to the amount of heat required to raise one pound of water one degree Fahr. We have seen that that number of foot pounds is 772 (Joule's mechanical equivalent of heat). Divide, then, by 772, the result is 72·2 units of heat, which, therefore, represents the work done during the generation of the steam. The only possible origin of this work is the heat absorbed during the process, which is generally called the latent heat, and amounts at the atmospheric pressure to 966·6 units. Subtracting, therefore, 72·2 from 966·6 we have a remainder of 894·4 units, which appears to be the actual latent heat, the heat necessary to the existence of the steam, since there seems to be no reason for making any further deduction. The work done during

the evaporation of one pound of water under higher pressure is considerably greater, while the so-called latent heat is less, and even the total heat, that is the sum of the latent and sensible heat, increases very little in comparison to the work. For instance at a pressure of 338·3 lbs. on the square inch the amount of work done is represented by 86·4 thermal units, and the total heat is 1236·2, the proportion of direct work to heat expended being about 1 to 14·3, instead of 1 to 16, as in the former case of steam generated at the atmospheric pressure. This fact is another argument in favour of using high-pressure steam, the economy of which is, however, established sufficiently by more obvious and forcible reasons, such as the consideration that higher rates of expansion can be employed, and that some sources of loss,—that by back pressure in the condenser, for instance,—are proportionate approximately to the *weight* of steam used rather than to the amount of work done.

In all practical cases, either of the *formation* or *condensation* of steam, the value ordinarily given as the latent heat of steam truly represents the amount of heat absorbed or liberated, as the case may be; and therefore this value is correctly used in estimating the amount of fuel required to generate, or water to condense, a given quantity of steam. The fact, however, that *only a portion of this heat is in reality latent*, the remaining portion being directly transformed into work, or from work into heat, in condensation, will throw some light on the action of steam in the cylinder of the engine.

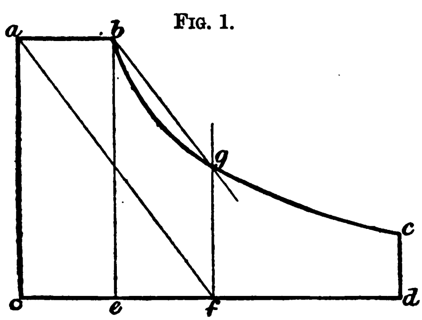
CHAPTER II.

THEORETICAL INDICATOR DIAGRAM.

The indicator diagram is, of course, the key to the action of the steam in the cylinder. A part of the work performed by the steam is spent in overcoming the friction of the engine itself, and consequently the efficiency of the *engine* is most fairly tested by the amount of external work absolutely performed against a brake or otherwise. Where the efficiency of the *steam* alone is concerned, however, the diagram is the only true criterion; and it will be necessary to deal with its theory carefully to prevent misunderstandings which are frequent in practice.

Fig. 1 represents an indicator diagram, not, indeed, actually taken from an engine, but drawn as a representative one, although differing in one or two respects from those obtained in practice. In the first place the horizontal line of exhaust, *o d*, is supposed to coincide with the zero of pressure, that is to say, the vacuum is

Fig. 1.



assumed to be perfect. The reason for this is that an inquiry into the action of expanding steam must deal with the whole of the work performed, and not with that part of it only which is utilized through the piston-rod. It has been already remarked that back pressure is a force which aids the strain in the piston-rod in opposing the advance of the piston, and for our present purposes it will not matter how much of the total opposing force is due to it. In every diagram the total amount of work done during one stroke is represented by the area, not of the figure usually taken, bounded below by the back-pressure line, but by one carried right down to the zero line; while another diagram taken simultaneously from the exhausting side of the piston, and bounded above and below by the back pressure and zero lines respectively, would represent the value of the wasted work.

The latter figure may be subtracted from the former by placing the one upon the other, with the zero lines in contact; the result being a diagram showing the amount of work which has been utilized. The diagram obtained in practice differs from such a one as this, in the fact that the figure deducted on account of back pressure from that representing the total work, is taken from the same side of the piston as the steam line, and at another time, whereas the opposing force must obviously act on the other side of the piston, and simultaneously with the pressure of the steam which it resists. In fact the back-pressure lines of the diagrams taken from the two ends of the cylinder should in

reality be exchanged, but in ordinary cases in which the diagrams are very similar, or when the values obtained from both are added together, the ultimate results are not affected. In finding the variation of strain upon the crank, however, during the stroke, it is most important to take the steam and back pressures which correspond in time, and to obtain these the back-pressure lines must be reversed. Further reference will be made to this below.

In the diagram shown in Fig. 1 the atmosphere line is omitted, the reason being that it is not wanted. It serves in ordinary cases partly as a starting point for measurement, in the absence of a zero line, which would serve the purpose better; and also, perhaps, partly to give a rough idea of the size without the necessity of using a scale, just as the figure of a man is used in an architectural drawing.

Few diagrams, except perhaps those taken from engines having Corliss, or some similar valve gear, approach at all in the sharpness of their corners to that shown in the figure. The effects of "*wire-drawing*," "*cushioning*," and "*lead*," will, however, be examined separately afterwards, and may, therefore, be left out of the question for the present.

THE WORK OF ADMISSION.—Let the diagram be divided into two portions by a vertical line, *b e*, drawn through the point *b*, at which steam is cut off. The rectangle *a b e o* represents the work done during the admission of the steam, or work done by "full steam" as it is sometimes called. If we disregard the slight

expansion of the steam which is necessary to its flow into the cylinder, we may say that up to the point of cut-off, it occupies as much space as it did before in the boiler, and is at the same pressure. It has not depreciated in value, therefore, by its change of place; and, consequently, it has produced of itself no work. It follows, then, that the work done during admission is not performed by the entering steam, which, in fact, acts merely as the water in an hydraulic engine does, as a flexible connection between the real origin of the work, and its point of application; the origin in this case being the steam which is formed in the boiler during the time of admission, or rather the heat which is expended in the process of ebullition. We have already seen, in considering the latent heat of steam, that a certain amount of work is done during its formation, and it is this work which finds its expression in the rectangular portion of the indicator diagram. Although it is not due, therefore, to the steam in the cylinder, yet it owes its existence to the formation, on the average, of an *equal amount* of steam in the boiler, and may, consequently, be regarded, without serious error, as belonging to the former, that is to say, it may be regarded as the first return for the heat expended during its production.

The value of the work done during admission, or during generation, which is the same thing, has been already calculated for one or two cases, and it is evident that it is always the product of the pressure and volume, or P V, when P represents the pressure of the

entering steam, and V its volume up to the point of cut-off. For instance, if P is 100 lbs. per square inch, and V is 1200 cubic inches, then the work done during admission, represented by the rectangle *a b e o*, is $1200 \times 100 = 120{,}000$ inch pounds, or 10,000 foot pounds.

THE WORK OF EXPANSION.—BOYLE'S LAW.—After the valve has closed, cutting off further communication with the boiler, the steam in the cylinder begins to do some work itself, by expansion, its pressure decreasing as it increases in volume. In order to find out how much work can be obtained in this way, as represented by the part of the diagram under the expansion line, it will be necessary to know what description of curve this is, and by what laws it is formed. The ordinary formulæ relating to expansion are based upon the following law, which sometimes bears the name of Boyle, and sometimes of Mariotte, a law which, as will be seen hereafter, must be taken with the caution that it is only approximately true for steam expanding under ordinary circumstances.

"*The pressure of a gas varies inversely as the volume*," or "*the product of the pressure and volume of a gas is always a constant*," *other conditions remaining unaltered.*

This may be stated mathematically thus:—

$$p\,v = a \text{ constant} = \text{P V}, \qquad [1]$$

where p and v represent the pressure and volume respectively at any point in the stroke after steam has been cut off. And if r be the ratio of expansion, that

is to say, the proportion of the terminal volume to the initial volume V, then the terminal pressure will be $\frac{P}{r}$. For instance, if steam of 100 lbs. initial pressure be expanded five times, then the terminal pressure will be 20 lbs.

The expansion curve of a theoretical diagram may be easily drawn by finding a few points on it by geometrical construction, or by calculating a few values of p from equation [1]. The geometrical method is very rapid and simple. In Fig. 1 through any point f in $e\,d$ draw a vertical line, join $a\,f$, and through b draw a line $b\,g$ parallel to $a\,f$, and cutting the vertical line through f in g. Then g is a point on the curve. A few points being found in this way, the curve must be drawn in by hand. It will be found to be a hyperbola, and anyone familiar with the properties of the conic sections will recognize equation [1] as that of a rectangular hyperbola referred to its asymptotes as axes.

The area of the expansion portion of the diagram $b\,c\,d\,e$ can now be found. Since

$$p = \frac{C}{v},$$

$$\textit{Area } b\,c\,d\,e = \int p\,d\,v = \int \frac{C\,d\,v}{v}$$

$$= C \log._e v.$$

Taking this between the limits V and r V, which are respectively the initial and terminal volumes, we have

$$\textit{Area } b\,c\,d\,e = C \log._e r;$$

and since by equation [1] C = P V,

$$\textit{Area } b\,c\,d\,e = P\,V \log._e r. \qquad [2]$$

Adding to this the area of the rectangular part of the diagram, we obtain

$$\textit{Total area of diagram} = P\,V\,(1 + \log._e r), \qquad [3]$$

which, when divided by the terminal volume $r\,V$, gives the mean or average pressure throughout the stroke;

$$Mean\ pressure = \frac{P}{r}\,(1 + \log._e r). \qquad [4]$$

This is the ordinary formula given in text-books on steam, and from which tables of mean pressures are calculated.

It is easy to see, as well from the diagram as from equations [3] and [4] or from the tables of mean pressures, that as the ratio of expansion increases the total work done also increases, but at a constantly slackening rate. Thus for an increase of r from 2 to 4, and another from 4 to 8, the increase of total work is in the proportion of 1·693 to 2·386 in the first case, but only 2·386 to 3·079 in the second. In fact, by doubling r we do not double the total work, nor, indeed, increase it in any fixed proportion, but in each case a fixed amount is *added* to it, namely, ·693 of the direct or full steam work. Hence consecutive additions of ·693 of the direct work are obtained at the cost of additions to the final volume of 1, 2, 4, 8, 16 etc. times the initial volume respectively.

A curve may be drawn showing the relations between the total work done and the ratio of expansion, by using the values of the latter as horizontal dimensions, as in the indicator diagram, while the vertical dimensions give the corresponding total amount of work, instead of pressure as in the ordinary diagram. These latter will be found from equation [3], and in drawing the curve it is convenient to use the same unit of measurement for the work as for the ratio of expansion, in which case the vertical dimension becomes simply

$1 + \log._e r$. The curve produced is shown in Fig. 3, marked *a*, and it corresponds to the expansion curve

Fig. 2.

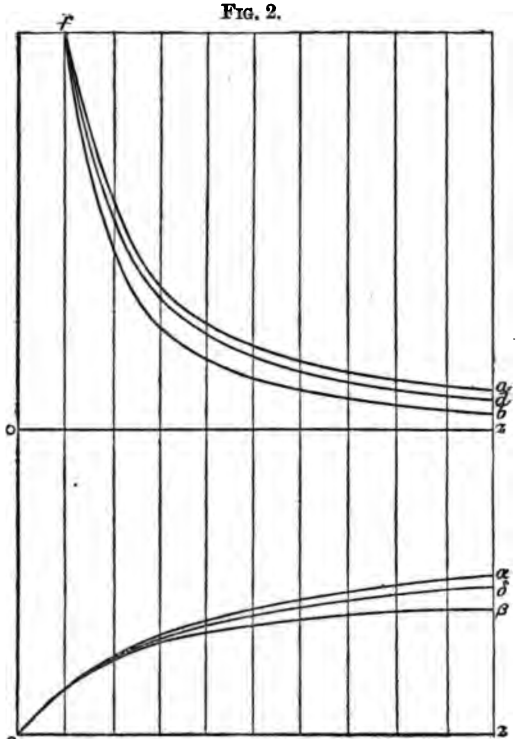


Fig. 3.

marked *a* in Fig. 2 above it, the values of the ratio of expansion coinciding in each. It may be called the curve of *work* or *efficiency*. From equation [3] it is evident that it rises continually, although with decreasing rapidity; and since $\log._e r$ may be increased to any extent by taking a sufficiently large value for *r*,

therefore the height to which the curve rises if continued far enough is unlimited;—in mathematical language the curve has no horizontal asymptote. In other words there is no limit to the amount of work obtainable from a fixed amount of steam, if only expansion be carried sufficiently far. This result, which is obviously absurd, since steam is not a creator, but only a vehicle of force, and therefore can convey a finite quantity only, follows irresistibly from Boyle's law of expansion. We are consequently led to the conclusion that for some reason the law is not strictly applicable to the case of steam expanding in the cylinder of an engine. The cause of this becomes apparent on a further examination of the law, the last clause of which stipulates that all the conditions of the case shall remain unchanged, except the volume and pressure. The temperature, therefore, must be constant. Hence the law applies only to a gas expanding without itself doing work, that is to a gas to which heat is constantly supplied from without, which heat is the source of the work performed. Where no such external source of work exists, the expanding gas must give up some of its own heat, and the diminution of temperature which results is necessarily accompanied by a corresponding diminution in volume, or rather by a less increase of volume than would otherwise take place. This is the case with steam under ordinary circumstances, and even when heat is supplied to steam during expansion, it is not sufficient to prevent entirely any lowering of the temperature.

CHAPTER III.

EFFECT OF COOLING DURING EXPANSION.

In finding out what allowance must be made for the effect of cooling during the expansion of steam under pressure, we shall have to suppose that no loss or gain of heat takes place from without; that is to say, that the sole cause of alteration of temperature is the work produced.

We will call v the volume, and τ the temperature (absolute) of a pound of steam of pressure p. We will suppose that the volume remains constant, while the pressure is reduced to $\frac{p}{2}$ by lowering the temperature to $\frac{\tau}{2}$. This will be effected by the removal of $\frac{\tau}{2} \times \cdot 37$ units of heat, $\cdot 37$ being the specific heat of steam at constant volume. Now let the pressure remain constant at $\frac{p}{2}$, and let the heat which has been removed be restored again, the volume being meanwhile allowed to increase. The augmentation of temperature will be

$$\frac{\frac{\tau}{2} \times \cdot 37}{\cdot 48},$$

since $\cdot 48$ is the specific heat of steam under constant pressure; and adding this to the temperature at the

end of the first part of the process, we obtain the final temperature,

$$\frac{\tau}{2} + \frac{\frac{\tau}{2} \times \cdot 37}{\cdot 48} = \frac{\tau}{2}\left(1 + \frac{1}{1\cdot 3}\right) = \cdot 88 \times \tau.$$

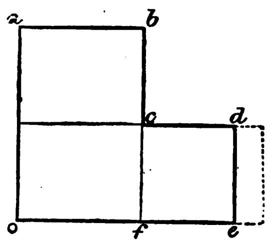
The alteration of volume will be proportional to that of the temperature during the second half of the operation, and the final volume is therefore

$$v\left(1 + \frac{1}{1\cdot 3}\right) = 1\cdot 77 \times v.$$

This shows a falling off of temperature of $\cdot 12$ of τ, and of volume of $\cdot 23$ of v, from what Boyle's law would have given.

The experiment just described may be represented by a diagram, as in Fig. 4, in which the area $c\,d\,e\,f$ indicates the work done during the expansion of the steam by the restoration of its heat, this amount of work requiring a certain reduction of temperature in the end; and, therefore, a smaller volume than would be in conformity with Boyle's law, as shown by the dotted lines. The conditions of this experiment are evidently different from any which occur in practice, since the variations of pressure and volume are here specially arranged to take place successively instead of simultaneously, the effect of this being the production of a smaller amount of work, and therefore not so great a reduction of temperature.

FIG. 4.



In every case, such as the above, whatever the expanding gas may be, if γ be the ratio of the specific heat at constant pressure to that at constant volume, then it is evident that

$$\gamma \frac{v_2 - v_1}{v_1} = \frac{p_1 - p_2}{p_2};$$

or, using the differential notation,

$$\gamma \frac{\Delta v}{v} = - \frac{\Delta p}{p + \Delta p}, \qquad [1]$$

Δp being a *decrement.*

And since during the first part of the process the decrease of τ is proportional to that of p, while in the second part the increase is $\frac{1}{\gamma}$ of the former decrease; therefore, if $\Delta \tau$ is the total decrement,

$$\frac{\Delta \tau}{\tau} = \left(1 - \frac{1}{\gamma}\right) \frac{\Delta p}{p}. \qquad [2]$$

Now in equation [1] let the increments diminish without limit, then

$$\gamma \frac{d v}{d p} = - \frac{v}{p},$$

from which we obtain by integration

$$\gamma \log._e v = \mathrm{C} - \log._e p,$$

$$v^{\gamma} = \mathrm{C} p^{-1},$$

or

$$v^{-\gamma} \propto p. \qquad [3]$$

From equation [2], when the increments diminish without limit,

$$\frac{d \tau}{d p} = \left(1 - \frac{1}{\gamma}\right) \frac{\tau}{p};$$

and integrating, as before,

$$\log._e \tau = \left(1 - \frac{1}{\gamma}\right) \log._e p + \mathrm{C},$$

$$\tau = \mathrm{C} p^{\left(1 - \frac{1}{\gamma}\right)}. \qquad [4]$$

These two equations, [3] and [4], represent, therefore, the relations between the pressure, volume, and temperature of gas undergoing expansion without loss or gain of heat from without.

For steam γ is $\cdot 48$ divided by $\cdot 37$ or $1 \cdot 3$, and equations [3] and [4] become

$$p \propto v^{-1 \cdot 3}. \qquad [5]$$

$$\tau \propto p^{\cdot 23}. \qquad [6]$$

It must be remembered that these equations hold good only on the condition that the steam expands as a fixed gas, that is to say, without condensation, a condition which is complied with in rare cases only, as will be shown immediately.

A diagram may be drawn from equation [5] by finding a few values of v by means of common logarithms, and setting off a few points on the curve as betore. The curve produced is shown in Fig. 2, marked b, the one marked a being the hyperbola obtained from equation [1], page 13, and representing expansion by Boyle's law. The divergence of the two curves at first is very considerable, but as they recede from o they gradually converge again. They never meet, however, and the interval between them decreases more slowly than the distance of either from the horizontal line of zero pressure $o\,z$, which is an asymptote to both. The zero of volume is also theoretically an asymptote to each curve at the other end, but this fact has no practical significance. For convenience in comparison, the two curves are drawn, intersecting at f, the unit of volume. Any arbitrary value may be given to the pressure at this point to suit the circumstances, since the equations employed are relative only.

The amount of work done, as represented by the area of a diagram drawn in conformity with equation [5], may now be found. The direct work will be P V, as before. For the expansion portion of the diagram, since

$$p = \mathrm{C}\,v^{-1\cdot3},$$

$$\int p\,d\,v = \mathrm{C}\int v^{-1\cdot3}\,d\,v$$

$$= \mathrm{C}\left(\frac{-\,v^{-\cdot3}}{\cdot3}\right).$$

Taking this between the limits V and r V, the initial and terminal volumes, we have

$$\text{Area required} = \left(\frac{1 - r^{-\cdot 3}}{\cdot 3}\right) \mathrm{C\,V}^{-\cdot 3}.$$

But since

$$\mathrm{C} = p v^{1\cdot 3},$$

therefore

$$\mathrm{C} = \mathrm{P\,V}^{1\cdot 3};$$

and substituting this value of C in the foregoing equation, we obtain

$$\text{Area required} = \mathrm{P\,V}\left(\frac{1 - r^{-\cdot 3}}{\cdot 3}\right);$$

and adding to this the rectangular area representing direct work,

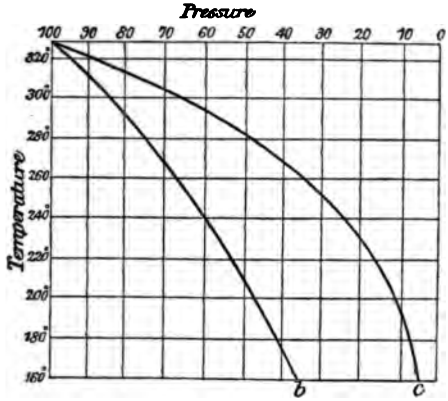
$$\text{Total area} = \mathrm{P\,V}\left(1 + \frac{1 - r^{-\cdot 3}}{\cdot 3}\right)$$

$$= \mathrm{P\,V}\left(\frac{1\cdot 3 - r^{-\cdot 3}}{\cdot 3}\right). \qquad [7]$$

This equation, which represents the relation of the total work done to the ratio of expansion, on the supposition of steam expanding as a fixed gas, may be embodied in a curve of efficiency, as shown in Fig. 3, marked β, similar to the one obtained from Boyle's law, marked a. It was stated before that the latter curve rose to an indefinite height, there being no limit to the value to which the total work might attain. This is not the case, however, with the value of the total work as given in equation [7]; for as r increases indefinitely, this value approaches to P V $\times$ 4·33 as a limit. Hence the curve of efficiency just obtained cannot rise above this amount; in other words, it has a horizontal asymptote at this height. This result appears, at least, nearer to the truth than the former, which is palpably absurd.

Condensation Produced by Expansion.—The relation of the temperature and pressure of steam, expanding as a fixed gas is given in equation [6] page 20. This also may be embodied in a curve, in the same manner as equations [5] and [7], although the form of the curve will be entirely different. It is shown in Fig. 5, marked *b*, the horizontal dimensions representing

Fig. 5.



pressure, and the vertical ones temperature, the zero of the former only being shown on the diagram. So far as this curve is concerned the initial temperature and pressure might both be arbitrary, since equation [6] is purely relative. They have been taken, however, at 790° Fahr. absolute, and 100 pounds on the square inch respectively, to facilitate comparison with another curve, marked *c* in Fig. 5, which represents the relation between the pressure of steam and its temperature of saturation, that is to say, the lowest temperature at

which it can exist, in fact the boiling point at any pressure. This curve is generally known as Regnault's curve, being the result of a long series of carefully-conducted experiments undertaken by M. Regnault at the request of the French Academy.

In Fig. 5, curve *b* falls very much more rapidly than curve *c*, which means that the theoretical temperature of steam expanding as a fixed gas falls during the process much more rapidly than does the temperature of saturation. Hence, if at the commencement the former is higher than the latter, it will not long remain so, and would ultimately fall below it, if that were possible. For instance, if steam of 100 lbs. pressure per square inch, at a temperature of 790° Fahr. (absolute) were expanded to a pressure of 50 lbs., the temperature which would result would be, according to the curve above found, 673°, whereas the boiling point at that pressure is 742°, which is 69° higher. It is therefore evident that *saturated steam in expanding requires more heat for transformation into work than that which it can liberate by a diminution of its own sensible heat, the possible amount of this diminution being limited to the extent indicated by Regnault's curve. In the absence of any external source of heat, the remainder of the heat required can only be supplied by the condensation of a portion of the expanding steam;* this condensation, which sets in immediately the temperature of saturation is reached, resulting in the liberation of latent heat to the extent needed. The work produced is therefore the effect of the expenditure both of sensible and of latent heat.

The fact of the partial condensation of saturated steam expanding without receiving heat from external sources is not dependent for proof upon theory only. The expansion may be accomplished in a glass cylinder, when a cloud or mist will be observed to form, which is due to condensation, since dry steam is invisible. This cloud is also seen over a locomotive funnel, or any jet of steam issuing under considerable pressure, when work is done by the expansion of the steam against the pressure of the atmosphere. If the pressure of the steam before escaping is comparatively low, so that very little expansion takes place, the steam is hardly, if at all, visible.

The mathematical relations between the volume and pressure of steam expanding continually at the temperature of saturation, and therefore constantly undergoing partial condensation, are very intricate, owing partly to the difficulty of expressing Regnault's curve accurately and briefly, and also to the variation of the latent heat of steam at various pressures. A full examination of the subject will be found in Professor Rankine's 'Manual of the Steam Engine,' § 281 *et seq.*, but practically the formulæ there obtained appear to be too cumbrous for ordinary use. In § 285, however, a very convenient and simple approximation is given, which is very nearly accurate when the initial pressure lies between one and twelve atmospheres, or from 15 to 180 lbs. This formula is

$$p \propto v^{-\frac{10}{9}},$$

and it may be solved in particular cases by means of common logarithms.

The diagram produced by using this equation for tracing the expansion line is shown in Fig. 2, curve *d*, which is drawn intersecting the two curves already found, namely *a* and *b*, at the point *f*, the initial pressure being arbitrary as before, provided that it falls between the limits named above.

This curve *d* forms the second and comparatively correct alteration of the expansion line from that given by Boyle's law, which we found to be applicable only in the case of a gas expanding without doing work; while the former alteration, *viz.* curve *b*, applied to a working gas, but not to a vapour which condenses during the process.

In comparing these three curves as shown in Fig. 2, the most noticeable fact is that while the curve *b*, for non-condensing steam, deviates considerably from the hyperbolic *a*, that for saturated steam *d* lies nearly midway between the other two, instead of still further from the hyperbolic, as might at first sight have appeared to be probable. This indicates that, *cæteris paribus*, steam condensing during expansion produces more work than the same volume of superheated steam expanding from the same initial pressure without condensing. *The condensation of steam in the cylinder is therefore not detrimental to economy, but the very reverse.* From one point of view the reason for this increase of work,—or of volume corresponding to any given pressure,—in the case of condensing steam appears to be that

the temperature is much higher, as already shown, than the theoretical temperature which would result if condensation did not intervene; and the larger volume due to this higher temperature more than compensates for the loss of volume by condensation, which loss it must be remembered is very small owing to the high latent heat of steam. *In short the result of condensation is that there is a little less weight of steam, but it is much hotter, and consequently more bulky, than it otherwise would be.*

Regarded from an economical stand-point the increase of efficiency may be accounted for by the fact that some of the latent heat of the steam becomes transmuted into mechanical work, instead of passing on unused to the condenser, and so being thrown away with the discharge water. All the heat which enters the condenser, whether sensible or latent, except a certain small percentage which is restored to the boiler in the feed water, is wasted so far as mechanical effect is concerned. It has only served to render the transformation of *other* heat into work possible, being one of the conditions of the working of the engine, like the stiffness of the piston-rod; and where steam is used without expansion, the latent heat residing in it serves solely to give to it the quality of resistance, in order that it may form a flexible connection between the source of work and its point of application, as shown above. Condensation in the cylinder is therefore a means of utilizing some of this heat necessarily employed, but which otherwise is ultimately all wasted. In either aspect of the matter

therefore the increase of efficiency shown by curve *d*, due to partial condensation, appears to be in harmony with what we should have expected.

Although this theory has been well established, almost contemporaneously by Clausius in Germany and by Rankine in England, it has been ignored or set aside by some recent writers on the steam-engine, whose works appear to us to have suffered materially in consequence. For instance, it is much to be regretted that Mr. P. Kaüffer's Treatise on 'Steam in the Engine, its Heat and its Work,'* in spite of the valuable suggestions it contains, supports the old theory. His argument against condensation taking place in the cylinder is that since heat is set free during the compression of steam, it follows that absorption rather than liberation of heat must take place during expansion. This argument needs only to be followed out to its legitimate issue to prove the fact of condensation in the cylinder; for it is because this absorption of heat for transformation into work is in excess of that supplied by the greatest possible decrease of temperature of the steam, that recourse is necessary to the store of latent heat in the steam, the abstraction of some of which reduces a corresponding amount of the steam to water. Probably the misapprehension arises in this case from the loose meaning commonly given to the term "*latent heat*," as already pointed out. This appears in the following sentence, where the reference evidently is to heat transformed from or into mechanical work, not to that which

* Blackie and Son, Glasgow and Edinburgh, 1873.

is an essential condition of change of state, as from solid to liquid. "Latent heat is set free when the molecules are brought nearer by mechanical compression, and heat is bound or transformed into latent heat during the reverse operation, which is expansion."

The efficiency of steam expanding at the temperature of saturation, and partially condensing during the process, may now be determined.

It has been remarked that the formula

$$p = Cv^{-\frac{10}{9}}$$

gives a rate of efficiency about midway between those given by Boyle's law and that for uncondensing steam. The mathematical expression for the area of the diagram is found as follows:

$$\begin{aligned}\text{Area of expansion part} &= \int p\,dv \\ &= \int Cv^{-\frac{10}{9}}\,dv \\ &= -9\,Cv^{-\frac{1}{9}}.\end{aligned}$$

Taking this between the limits V and r V, and for C substituting, as before, its equivalent $PV^{\frac{10}{9}}$, we obtain

$$\text{Area of expansion part} = 9\,PV\left(1 - r^{-\frac{1}{9}}\right).$$

Let the area of the rectangular portion of the diagram now be added, then

$$\text{Total area} = PV\left(10 - \frac{9}{\sqrt[9]{r}}\right).$$

From this equation a curve may be drawn to represent the efficiency or work done for any given rate of expansion, by means of common logarithms as before. It is shown in Fig. 3, marked δ, and lies between those marked a and β in the same manner that the line d in Fig. 2 does between a and b. In the equation it is

evident that when r increases without limit, the value of the total area approaches to a limiting value of 10 P V; hence the curve δ has a horizontal asymptote at the height which represents this value. These limiting values or horizontal asymptotes to the curves β and δ are interesting as showing the relations of these curves to each other, and to curve a, which has no asymptote. It is very important, however, to remember that neither practically nor theoretically do they represent the limit to the amount of work obtainable from the steam, since the formulæ from which the first curve β is deduced apply to steam expanding for a short time only, and even then not with absolute accuracy; while the formula which forms the basis of curve δ is only a near approximation, and deviates very considerably from the truth in extreme cases, either of very high or very low pressure, or of exceptionally high rates of expansion.

It is not worth while to have recourse to long and cumbersome formulæ in order to attain an amount of accuracy which is lost again by the unavoidable uncertainties and errors in the data needful for applying them in practice; and consequently even the above approximations will hardly supersede those based on Boyle's law and the hyperbolic curve for ordinary purposes. A more general knowledge, however, if not of these formulæ themselves, at least of the principles on which they are based, might prevent many errors in practice as well as in theory. The most fruitful sour[illegible] errors lies in the application of a formul[illegible] true in certain ordinary cases to a[illegible]

which its deviation from the truth is perhaps also extreme. Thus the error resulting from the use of Boyle's law in determining the mean or terminal pressure in a cylinder may be small as compared with those resulting from uncertainty of the true rate of expansion, owing to wire-drawing, steam enclosed in passages, clearance, &c. But when the law is used as the basis of a calculation of the economy of using very high pressures and extreme rates of expansion, the error is by no means insignificant, and it is one against which practical men should be on their guard; for even in a paper on this subject read by Professor Osborne Reynolds to the Manchester Association of Employers, Foremen Engineers, and Draughtsmen, the value of some otherwise useful and carefully-prepared Tables is seriously impaired by their being based upon this law. These Tables are headed by the remark that the cylinders are supposed to be kept dry by a steam jacket or otherwise: and since the ratio of expansion is found by dividing the initial by the final pressure, it follows that the steam is supposed to retain its initial temperature, in other words to do no work in expansion. In this case the amount of steam used [illegible] cylinder would be [illegible] criterion of the consu[illegible]l, since it is [illegible] sole [illegible] work. [illegible] and, if the [illegible] the [illegible] e work, th[illegible] [illegible] approxima[illegible] [illegible] m being [illegible] the Table[illegible] [illegible] l of 5 lbs. are [illegible]

give a ratio of expansion of 61, while the formula at page 25, for steam expanding under these circumstances, which may be taken as at least more correct than Boyle's law, gives only 40·4. The extent of the error in the efficiency is not so great as the error in the ratio of expansion, but it will be found to amount to about 20 per cent. From the head note to the Tables it would appear they were framed on neither of these suppositions alone, but on both combined, Boyle's law being taken as nearly true of steam expanding at the temperature of saturation, but kept from condensation by a steam jacket, and the consumption of steam in the jacket being neglected as insignificant. By this means the error is merely so divided as to escape notice; its total amount remains substantially the same. The subject will be further pursued in dealing with the steam jacket.

CHAPTER IV.

THE STEAM JACKET.

It has been already shown that the condensation of a certain portion of the steam expanding in a cylinder results in a greater volume corresponding to any given pressure than would be the case if expansion took place without condensation, other circumstances being the same. Hence it might appear to follow that the prevention of condensation would be detrimental to economy. Steam jacketing has this very object in view, namely, the prevention of condensation in the cylinder by supplying heat to the expanding steam.

This subject has two aspects, the geometrical aspect presented by the diagram, and the economical aspect. Looked at from either of these points of view the result is the same.

In the first place, referring to the expansion curves shown in Fig. 2, the effect of condensation was seen to be the alteration from curve *b* to curve *d*, that is to say an increase of efficiency, which was explained on the ground that the increase of temperature more than compensated for the decrease of weight. Where a steam jacket is used the increased temperature, that is the saturation temperature, is retained, *without* decrease of weight. Hence the volume corresponding to any given pressure is greater, and the amount of work done also greater in

D

this case than in that of condensing steam, and the expansion curve produced with a steam jacket should therefore lie between curve *b* and the hyperbola, *a*. Its position is given approximately by the formula

$$p \propto v^{-\frac{17}{16}},$$

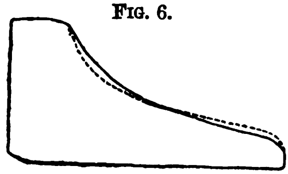
it being assumed that condensation is completely prevented. This appears to be usually the case where the steam jacket is properly employed, since wet steam is a rapid absorber of heat, while dry steam is a very slow one, consequently no great departure from the temperature of saturation takes place on one side or the other. It cannot be expected, however, that in cylinders of great diameter or with a very rapid stroke the effects of the steam jacket will reach the middle of the cylinder quickly enough to prevent condensation.

From the other—the economical—point of view the result of the jacket is to obtain work from steam which does not enter the cylinder, by condensing in the jacket instead of in the cylinder, it being remembered that this condensation is due to the transformation of latent heat into mechanical energy. The amount of work done in proportion to the steam used in the cylinder, is certainly increased by this means; but it does not follow that there has been an increase of efficiency, since the expenditure of steam is now not only that used in the cylinder, but also that which is condensed in the jacket. The direct result therefore is to send more uncondensed steam, and so more latent heat, to the condenser at each stroke. On these theoretical grounds

therefore it would be difficult to show any appreciable increase of economy resulting from the jacket, although it might be very useful to increase the power of an engine which was too small for its work without altering its rate of expansion.

The real advantage of the steam jacket must be sought for in the fact that the condensation in the cylinder which it is intended to prevent is indirectly a great source of loss. A cloud or mist is produced, which is thickest at the end of the stroke, and during the exhaust. This removes heat from the cylinder partly perhaps by direct conduction, but chiefly by settling as dew on the surfaces during the stroke and evaporating again during the exhaust, when the pressure is reduced. The latent heat taken up during this evaporation is borrowed from the metal of the cylinder, and must be repaid by the steam which enters for the next stroke, and which can ill afford to be thus cooled at the outset. A certain amount of alternation of temperature is a necessary consequence of expanding under ordinary circumstances, and any cooling of the metal of the cylinder which takes place *during the stroke*, adds, by heating the steam, to the small end of the diagram; while the reheating at the commencement of each stroke takes away an equal portion from the other end. The dotted line in Fig. 6 represents this action, and it is evident that no great loss results so far. But any cooling of the

FIG. 6.



cylinder which takes place during the exhaust requires the subtraction from the beginning of the diagram without any corresponding compensation at the end, and this cooling is greatly assisted by the presence of moisture. It will vary in extent in different cases, being very slight where the terminal pressure in the cylinder is the same as the back pressure in the condenser, and increasing as the difference between these increases. Where a steam jacket is used, and condensation during expansion consequently reduced to almost nil, the only source of loss of heat during the exhaust is by conduction and radiation, and it is no doubt not very great, although reheating of the metal of the cylinder commences probably very soon after the exhaust opens.

The use of the steam jacket has been somewhat extended of late. It was originally applied to the body of the cylinder only; then to the end and cover, and finally some engineers have admitted steam to the piston. This is, of course, expensive, and involves extra joints; but it no doubt tends to economy appreciably where the surfaces are large. It seems probable, however, that more advantage accrues from the steaming of one square foot of the working barrel of a cylinder, than of an equal area of cover or piston, since the former is always kept comparatively clean by the friction of the piston; while the latter surfaces soon become coated with a black carbonaceous deposit, the product of the partial decomposition of the lubricants, which prevents the passage of heat to the steam in the cylinder just as in a familiar experiment a similar

deposit on the bottom of a kettle protects the hand on which it rests.

With regard to hot-air jackets little need be said, as they are not extensively used. Steam is a more convenient and economical carrier of heat than a fixed gas, and a supply is always at hand; hence it is almost invariably employed. The practice of jacketing with exhaust steam is happily now almost entirely abandoned, and it is surprising that anyone should have expected that cold steam would give up heat to warmer steam. It is also objectionable to supply the jacket with steam which is on its way to the cylinder, the result being to condense the steam partially *before* instead of *during* expansion. The steam for the jacket may be taken from the steam chest, or steam pipe, but should not be returned: the condensed water is blown out occasionally.

CHAPTER V.

SUPERHEATING.

THE object of the steam jacket is to supply heat to the expanding steam, in order to prevent its partial condensation. Another way of attaining the same result, to some extent at any rate, is to impart an extra supply of heat to the steam before it enters the cylinder, and this is called *superheating*, since the temperature is raised above that of saturation. The extra supply of heat is sensible heat, and it produces an increase in the volume of the steam proportionate to the increase of the absolute temperature, since the pressure is not altered in the process.

It has been shown above that the expansion of steam as a fixed gas, which may be taken as the state of superheated steam, is represented by the equation

$$p = Cv^{-1\cdot 3},$$

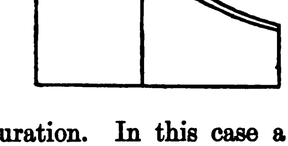
and the resulting expansion line is marked b in Fig. 2. Also it was shown that the temperature under these circumstances follows the law

$$\tau = Cp^{\cdot 23},$$

and the corresponding curve is shown at b, Fig. 5. From this latter equation and curve it appears that the fall of temperature is so rapid, that the point of saturation is reached after a comparatively short period of

expansion, and from that time partial condensation takes place unless prevented by the use of a steam jacket. Consequently under these conditions expansion commences by curve *b*, Fig. 2, and then after a time, the length of which depends of course on the extent of superheating, continues by a curve similar to *d*. Now, as the curve *b* lies *within* the curve *d*, when starting from a common point, it follows that a given volume of superheated steam would produce in expanding a diagram *within* and smaller than that which the same volume of saturated steam would give, and the excess of the latter over the former would be greater the greater the degree of superheating. This is illustrated in Fig. 7, where the fuller expansion line is that of saturated steam, expanding, without loss or gain of heat, from a pressure of 100 lbs. on the square inch; while the other curve represents the expansion of the same volume of superheated steam, the initial temperature of which is 400° Fahr., or 91° above the temperature of saturation. In this case a pressure of 53 lbs. is reached before condensation begins, but from that point the expansion follows the law for saturated steam, and the pressure maintains a fixed relation to that of the steam which was not superheated, at any given volume.

FIG. 7.



Volume for volume therefore saturated steam is worth

more than superheated, at the same pressure, but in comparing equal weights the advantage lies obviously on the other side, both because superheated steam contains in itself more heat, and also because direct work is done in the process of superheating. The whole of the extra heat so expended may be transformed into work, by working with a rather higher ratio of expansion than would otherwise be used, in order to avoid an increase of the terminal pressure and temperature of the steam; and under these conditions a saving is certainly the result; but it is not a very great saving, owing to the obstacles which in practice prevent superheating from being carried beyond certain limits. The chief object of superheating appears, in fact, to be not so much any direct economy which may result, as the prevention of one great cause of inefficiency, namely, the use of wet steam or vapour. Long steam-pipes, or want of lagging, or a slight but continuous priming of the boiler, are some of the causes of the steam entering the cylinder in a moist state; and the remarks made above on the loss resulting from condensation during expansion apply here with still greater force, for in this case the moisture is owing to waste or deficiency of latent heat, instead of to its transformation into work. The water serves both as a vehicle of its own heat from the boiler to the condenser, and also takes up even more heat by evaporating from the surface of the cylinder during the exhaust.

On the other hand, a little moisture in steam is said to serve as a lubricant to the working surfaces,

and several cases have occurred in which superheating has resulted in a great amount of friction and wear of valve-faces and cylinders.

Again, for superheating which goes further than drying, and indeed for drying only at high pressures, a temperature is required which chars the packing in the glands and the oil or tallow used as a lubricant. This difficulty as regards the glands may be overcome apparently by the use of metallic packing; and for the other matter the use of lubricants in cylinders and steam chests is beginning to go out of practice, owing to the verdigris formed in the surface condenser, which when returned into the boiler produces severe corrosion.

Superheating then should aim at hardly drying the steam supplied to the engines where a surface condenser is used; and with this precaution it appears probable that lubrication might be dispensed with in almost all cases.

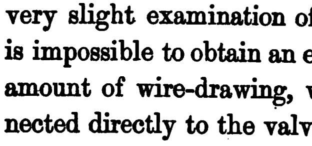
The term "*combined steam*" has been applied to slightly superheated steam resulting from a mixture in the steam chest of superheated and saturated steam arriving by different passages, but from the same boiler. The only apparent advantage of this arrangement is that it enables the degree of superheating to be easily and accurately regulated, and so the state above recommended, namely that just short if anything of dryness, may be obtained.

CHAPTER VI.

THROTTLING AND WIRE-DRAWING.

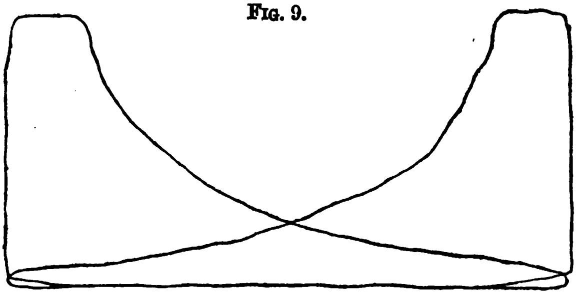
When steam is reduced in pressure by passing through a contracted passage, as in a stop-valve partly closed, or in the common "throttle-valve," it is said to be "*throttled.*" The term "*wire-drawing*" is almost identical in meaning with throttling, but refers especially to the slow cutting off of steam by an ordinary slide-valve, the result in the diagram being a gradual slanting downwards of the steam line until it passes imperceptibly into the expansion line. Fig. 8 is an example of this, and the dotted lines show the effect of a quick cut-off obtained by means of an expansion valve. A very slight examination of valve gearing shows that it is impossible to obtain an early cut-off without a certain amount of wire-drawing, when a single eccentric connected directly to the valve spindle is used. If under these circumstances an earlier cut-off than at half stroke is attempted, the wire-drawing becomes excessive. Numberless forms of valve gear have been arranged to avoid wire-drawing with high rates of expansion, the commonest and simplest being by means of double

Fig. 8.



eccentrics and slides, gridirons, &c., which may be designed to give a sufficiently rapid and early cut-off for ordinary purposes. There is another class of valve gear represented by the Corliss, which is in fact an arrangement of steam cocks opened and closed quite

FIG. 9.



suddenly by means of steam springs. Fig. 9 is a diagram taken from an engine fitted with Corliss gear, and it shows a perfectly steady steam line for one-eighth of the stroke, and then a sharp cut-off, with expansion through the remaining seven-eighths of the stroke.

The effects of wire-drawing and throttling are more complicated than would appear at first sight. A common but unsatisfactory argument against both is that such expansion is unaccompanied by work, and is therefore wasteful. Where no work is performed, no expenditure of heat will take place, and hence we should infer that the temperature of steam would be the same after expansion, in such a case, as it was before. This has been proved to be very nearly true by experiments in which the steam was allowed to expand through a porous

plug—a clear case of throttling. The slight loss of temperature which takes place is greater the more moisture there is present; and it may be accounted for on the supposition that the friction of the steam against the passages results partly in electricity, which is one form of force, and therefore is produced at the expense of some other form; in this case heat. Electricity is a too favourite resource for accounting for unexpected phenomena, but in the present case the hypothesis is supported by some amount of proof, since the friction of steam escaping through tortuous and difficult passages formed in boxwood, has been used successfully for the production of electricity in considerable quantities; indeed, a machine constructed upon this principle appeared many years ago in one of the exhibitions, but it was never put to any practical use owing to its want of economy. A description of it will be found in books on electricity.

Neglecting for a moment the slight loss of heat which occurs during throttling, and assuming that the temperature remains unchanged, we are led to the conclusion that *saturated* steam becomes *superheated* during the process; not indeed by any addition of heat, but by the reduction of pressure and consequent reduction of the temperature, which corresponds to saturation. In practice, the fall of temperature of the steam is not so great as the fall of the temperature of saturation, and consequently superheating does actually result to a greater or less extent according to circumstances. The

advantage of this must be admitted when the wastefulness of wet steam is remembered, but it hardly compensates for the loss which we shall find to accompany it in practice.

If absolutely no work were performed by steam during the process of throttling, no alteration would take place in the amount of work obtainable from it, and therefore there would be no depreciation of its value. It would be a mistake, however, to infer that the slight loss above referred to, which occurs during the operation, is the whole loss which results. We have already seen that of the whole work developed in the cylinder of an engine a part only acts on the piston-rod, while the rest is expended against the back pressure, and is therefore to all intents and purposes wasted. This waste in engines equal in other respects, namely, in total work, ratio of expansion, and back pressure, will be in proportion to the volume of steam used; and therefore although the work obtainable from throttled steam may be equal to that from unthrottled, the amount wasted, being in proportion to the volume used, will be greater in the former case than in the latter. For example, 2 cubic feet of steam at 30 lbs. pressure, are worth as much as 1 cubic foot at 60 lbs., the temperature being the same. If these two volumes of steam are allowed to expand behind pistons whose areas are in the same proportion, *viz.* as 2 to 1, the absolute mean pressures resulting from any given rate of expansion will maintain the proportion of the initial

pressures. Let them be 16 lbs. and 32 lbs. respectively; then subtracting from each the back pressure of say 4 lbs., we obtain the *mean effective pressures* in the two cases equal to 12 lbs. and 28 lbs.; and the work utilized on the piston-rods in the proportion of 24 to 28, or 6 to 7. The loss in this case therefore is $\frac{1}{7}$ of the whole.

The inference is that throttling and wire-drawing are accompanied by direct loss due to the slight cooling which takes place during the process, and by indirect waste owing to the increased proportion of work expended in overcoming the back pressure.

CHAPTER VII.

INTERMEDIATE EXPANSION IN COMPOUND ENGINES.

Cause of Intermediate Expansion.—The foregoing remarks have applied entirely to the expansion of steam in the cylinder of an engine; an important and rather peculiar case now presents itself of expansion of steam in leaving the high-pressure cylinder of a compound engine. This can be examined most simply in the case of an engine having an intermediate "*receiver*" into which the small cylinder exhausts, and from which the low-pressure cylinder draws its steam. It will be convenient for the present to neglect the size of this receiver, regarding it as very large comparatively to the cylinders, so that the fluctuations due to the alternate entrance and exit of steam may be disregarded.

At each stroke of the engine the high-pressure cylinder discharges a volume of steam equal to its own capacity into the receiver. During the same period there is drawn out from the receiver a volume sufficient to fill the low-pressure cylinder up to the point of cut-off. It is evident that if these two volumes, namely, the volume of the steam sent in and the volume of that drawn out of the receiver at each stroke, are the same, then the pressure at which the steam enters and leaves will be the same also. But if the outgoing volume is the greater a corresponding fall of pressure

must take place which will be roughly in inverse proportion to the increase of volume. Briefly then, and not strictly accurately, *terminal high* is to *initial low* as *capacity of low cut-off* is to *capacity of high.* For example, if the cylinders are in the proportion of 4 to 1, and if the low cuts-off at half stroke, then there will be a fall of pressure between the cylinders in the proportion of $4 \times \frac{1}{2} = 2$ to 1.

EFFECTS OF INTERMEDIATE EXPANSION.—This is the phenomenon of intermediate expansion and its direct cause; the next point will be to discover as far as possible its effects, and to find by what law it is regulated. The case just given in which the capacity of the low cut-off is twice that of the high, intermediate expansion being therefore in the ratio of 1 to 2, will serve very well as an example. The size of the receiver must still be disregarded. The work done during the admission of steam to the large cylinder will be equal to the product of the pressure in the receiver, multiplied by the capacity of that cylinder up to the cut-off, which is equal to twice the capacity of the high. It has been shown previously that work done during admission is not done by the entering steam (except in a few cases, of which the present is not one); therefore the source of this work in the low-pressure cylinder must be sought for at some earlier stage. Let the high cylinder be just opening to exhaust; the steam contained in it has a pressure double of that in the receiver, and it now expands suddenly down to the lower pressure of this steam, which it drives back through a space equal to its

own previous volume. The work done in this expansion is the product of the high capacity multiplied by the pressure in the receiver, or one-half of that done during admission into the low. As the piston of the high-pressure cylinder advances it drives out before it the remainder of the exhaust steam, which still fills the small cylinder after its sudden expansion. The work done during this process is again equal to the product of the capacity of the small cylinder multiplied by the pressure in the receiver, and it therefore completes the amount of work required to account for that done during admission into the low. This second portion is conveyed directly from the high piston to the low, the steam acting simply as a flexible connection between them; and it is the part of the work done in the high cylinder which does not pass into the piston-rod, and which is represented on the diagram by the area under the exhaust line, as before remarked, see page 10. The conclusion then is that in expanding between the cylinders the steam does a certain amount of work, equal to the product of the receiver-pressure multiplied by the difference of volume before and after expansion, which is the capacity of the low cut-off *minus* the capacity of the high cylinder. This work will be represented by a rectangular diagram. It is at first sight difficult to understand how steam expanding from a higher to a lower pressure should produce work against the latter only, instead of against a varying pressure as is the case in the cylinder. It is a constant pressure however which opposes itself to the expansion

of the steam. The explanation of the difficulty appears to be as follows:—If a small portion of the steam be considered apart from the rest it is clear that its expansion is more or less impeded by the inertia of the surrounding mass. Hence a gradual though extremely rapid change of pressure is the result, and the excess of pressure above that of the receiver is expended in overcoming the inertia of the steam set in motion. The total amount of this excess of work, being equal to the *vis viva* of the whole mass of steam set in motion, determines the velocity of the flow of the steam. And when equilibrium is restored and the steam comes to rest again under the reduced pressure, the *vis viva* is transformed back into heat, and consequently the only permanent work done is that of driving back the steam in the receiver at constant pressure which is represented by the rectangular diagram.

It is evident that steam expanding under these conditions will not obey precisely the same laws as when it expands against varying pressure in the cylinder. It is not worth while, however, to do more than state the relations which exist between the pressure, volume, and temperature on the supposition of steam being a perfect gas. They are very easily arrived at,—

$$\frac{v_2}{v_1} = 1 + \frac{p_1 - p_2}{p_2 \gamma};$$

$$\frac{\tau_2}{\tau_1} = \frac{p_2}{p_1}\left(1 + \frac{p_1 - p}{p_2 \gamma}\right).$$

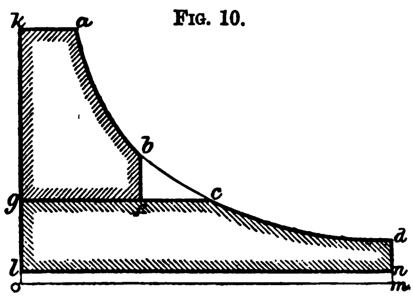
From this last formula it is easily shown that condensation almost as copious as that which takes place in

the cylinder must occur in this case also; and from this we may infer that a receiver should if possible be provided with some heating arrangement, such as a steam or hot-air jacket. In practice the external surface of the cylinder jacket serves usually to warm the receiver; and in some cases the receiver is a separate vessel, heated directly by the flues from the furnaces. In the latter arrangement there is probably more or less superheating in the receiver, but where a steam jacket only is employed the temperature will hardly be much in excess of the temperature of saturation, and consequently the relation of pressure to volume will be very nearly as before,

$$p \propto v^{-\frac{17}{16}}.$$

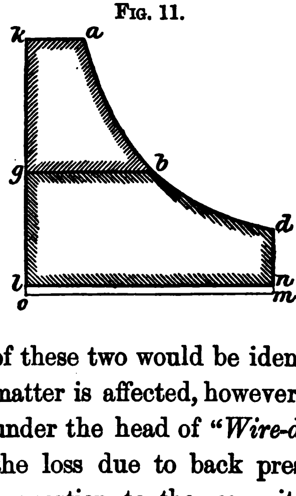
The question now arises, as to what amount of loss, if any, is due to intermediate expansion. Fig. 10 is a theoretical diagram. In order to avoid any variations of the curve due to the differing conditions of expansion in a compound engine, a steam jacket may be supposed to be applied throughout. Let the first part of the curve, *a b*, represent expansion in the smaller cylinder; *b c*, the intermediate expansion or drop in the receiver; and *c d* the expansion in the low-pressure cylinder. Then *a b f g k* will be

Fig. 10.



the high-pressure diagram; *g c d m o*, the low; and the waste which has resulted from having an intermediate drop is shown by the triangle *b f c*. *If therefore a drop can be avoided without altering the total ratio of expansion a saving to this extent will be effected. When, however, the only convenient mode of avoiding a drop would be to decrease the capacity of the large cylinder, and therefore also to diminish the total ratio of expansion, there would be no saving*; since more area is cut off from the end of the diagram than is saved in the middle, and the result is Fig. 11. The values of the low-pressure diagram are very nearly the same in each case; in fact if expansion were by Boyle's law they would be exactly the same; for the initial, and therefore also the mean pressure in the low would be in inverse proportion to the capacity, and therefore the product of these two would be identical in each case. Here the matter is affected, however, by the fact remarked upon under the head of "*Wire-drawing and Throttling*," that the loss due to back pressure in the condenser is in proportion to the capacity of the cylinder which exhausts into it. Thus if the choice of mean pressures is between 20 lbs. on a small piston, or 10 lbs. on one of double the size, and if the back pressure is 4 lbs., then the former of these gives just one-third more available

FIG. 11.



work than the latter. The area below the line $l\,n$ in Figs. 10 and 11 shows the amount of loss in each case due to back pressure. While this area increases with any increase of capacity of the low cylinder, the area of the high-pressure diagram increases also by the lowering of the line $g\,f\,c$, and the best result will be therefore attained when this line $g\,f\,c$ is brought down just so far that any further reduction would take more from the low-pressure diagram than it would add to the high. Where an expansion valve is used on the other hand, and intermediate expansion therefore prevented, the low-pressure cylinder may be made of such a capacity that the pressure of steam in it at the end of the stroke shall be hardly if at all higher than that in the condenser.

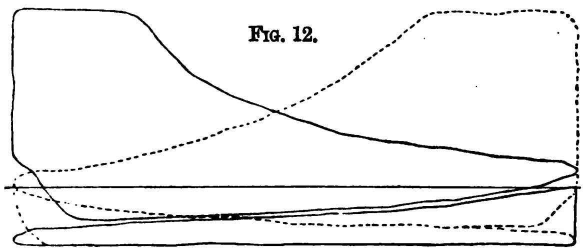
AVOIDANCE OF INTERMEDIATE EXPANSION.—There are several arrangements in use by which intermediate drop may be avoided altogether, or reduced to any desired extent, without diminishing the amount of expansion which takes place after the steam leaves the small cylinder. The commonest of these is that already referred to of providing the large cylinder with an expansion valve, by which means its capacity up to the point of cut-off may be reduced to that of the high cylinder. The practical objections to this arrangement are: *first*, the extra expense; *secondly*, the extra power absorbed in driving an expansion valve; and *thirdly*, the extra strength of rods and size of bearings necessary on account of the increased strain on the low-pressure crank produced by a comparatively high initial pressure

acting on a large area of piston. This last objection is partly met by the fact that low-pressure rods are usually made sufficiently strong to stand the high-pressure steam used in starting; but only partly, because the use of an expansion valve, if it does not lead to the breaking of rods, often results in the heating of the crank-pin or bearings, the weight of the large piston helping also to bring this about. Again, an ordinary slide-valve may easily be relieved of the pressure of the steam on its back, but where an expansion valve is used either no relieving is attempted, or it is but partially effected at great expense. This adds force therefore to either the first or second objection, and when the case is fully weighed, and the insignificance of the saving effected is remembered, it is hardly to be wondered at that low-pressure expansion slides are not more commonly used.

Another way of avoiding a drop of pressure is to make the pistons begin and end the stroke together, and to exhaust directly from the high-pressure cylinder into the low. In this class of engines the intermediate receiver is done away with, and the passages by which the steam exhausts from the one cylinder to the other are made as small as possible, one cylinder being even placed sometimes *within* the other. When the exhaust port of the high-pressure cylinder opens, the low piston is at the end of its stroke, so that no expansion of the exhaust steam from the high can take place except into the clearance of the low and the intermediate passages. As the two pistons advance, which they do simultaneously, the steam flows from the smaller to the larger

cylinder, expanding meanwhile. The communication between the cylinders is not closed until the end of the stroke, or nearly so, and consequently the lowest pressure of the exhaust in the high is the same as the terminal pressure in the low. Fig. 12 is a copy of cards

FIG. 12.



taken from engines of this class, and the coincidence of the exhaust line of the high diagram with the steam line of the low shows the reduction of pressure of the high exhaust referred to. The consequence of this reduction is that the high-pressure cylinder is subjected to the cooling influence of a pressure very little above that in the condenser; but the loss on this account is very slight indeed, if there is any, because it occurs only at the end of the exhaust stroke, and also because the second cylinder acts as a trap for any heat which would otherwise escape by this means to the condenser. The real practical objection to this description of engine is one which applies more to marine than to land engines; it is that the pistons must begin and end the stroke together, moving therefore always in the same, or always in opposite directions, so that where the cylinders are parallel and only two are used the *dead points* coincide.

To get over this difficulty some engineers have made a compromise, keeping the cylinders parallel, but the cranks some 20° or so out of the straight line, that is to say at an angle of about 160° with each other. By this means the engines go over the dead points without difficulty, and the pistons move very nearly together. The high-pressure piston ought to commence its stroke just before the other (and therefore the *low crank* should lead); then the only effect of the alteration is to give a higher back pressure against the small piston at the beginning of each stroke by compression of the exhaust steam until the low valve opens. This valve must be arranged to close again by the time that the high piston reaches the end of its stroke,—cutting off, that is to say at about three-quarters of the stroke of its own cylinder.

Some engines which have in other respects been built on this principle, have been so contrived that the *low piston* commences its stroke first; the result being a rush of steam into the large cylinder after the stroke has begun, and a sudden drop is thus brought about. The exhaust steam is also cushioned in the high cylinder after the low ceases to take steam. This would hardly be a disadvantage, but that it occurs just as the engines are going over the centres. The neatness of the diagrams of these engines is very much impaired by the irregularity of the low-pressure steam line, which begins low, but rises suddenly early in the stroke and then falls away gradually.

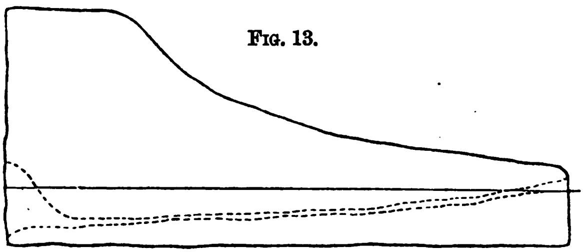
One curious form of continuous-expansion compound

engines is constructed somewhat on the principle of the bucket-and-plunger pump. One cylinder only is used, and the efficient area of the piston is reduced on one side to, say, one-half or one-third of its total area by means of a trunk, the other side of the piston having its whole surface exposed to pressure. The steam from the boiler is admitted on the reduced, or trunk, side, and it expands here, as in an ordinary high-pressure cylinder, to the end of the stroke. It exhausts, however, by an appropriate valve, to the other side of the piston, where it acts on a greater area and produces the return stroke, expanding ultimately to the whole capacity of the cylinder, and then exhausting into the condenser. The same cylinder is thus exposed to the highest and lowest pressure, *viz.* that of the entering steam and that of the condenser; so that one of the alleged advantages of compound engines is here sacrificed. It is noticeable, too, that the high-pressure steam is opposed only by the back pressure in the condenser, while the low-pressure steam during the return stroke is opposed by steam of the same pressure, the same steam in fact, acting, however, on a smaller area. In each case the atmospheric pressure on the trunk is in the same direction, assisting the high-pressure steam and opposing the low-pressure to an exactly equal extent. It follows therefore that the pressure during the return stroke must be more than that of the atmosphere, unless the latter is counterbalanced by a weight, or removed by the substitution of the condenser pressure. It is not easy to resort to this last expedient in the engines just described, except in a par-

tial manner, by using the outer end of the trunk as the ram of the air-pump. It is, however, resorted to in some engines identical in principle with these, though differing a little in form, the arrangement being something of this kind;—a high and a low pressure cylinder are placed in one line, say, for instance, the high above the low, and the pistons secured to a single piston-rod. The ends of the two cylinders which are next each other, that is, the bottom of the high and the top of the low, are always in free communication with each other, and it is from this space that the atmospheric pressure is removed by connection with the condenser. Steam from the boiler is admitted above the small piston, and completes a stroke, as before, in the high cylinder. On exhausting it passes to the under side of the large piston, and produces the up-stroke by pressure on increased area. Here the high-pressure steam is opposed by the pressure in the condenser, and the low-pressure by steam of equal pressure, as in the case of the trunk compound engine. The diagrams taken from these engines are very noteworthy; in fact, they appear to contradict the above remarks, by showing a back-pressure line in the high-pressure diagram coinciding more or less nearly with the steam line of the low, as in Fig. 12. The cause of the apparent contradiction is explained when the caution given in the second chapter is remembered, namely, that to obtain the true diagram from any cylinder, an exchange must be made of the two back-pressure lines from the opposite sides of the piston. The high-pressure diagram so corrected is shown by the full

lines in Fig. 13, but the low-pressure diagram either disappears altogether, in consequence of the coincidence of the steam and the back-pressure lines, or else becomes negative, as shown by the dotted lines in Fig. 13, in

FIG. 13.



consequence of the latter being the higher of the two. This is the graphic representation of the above-mentioned fact, that motion during the low-pressure stroke is produced, not by a higher pressure per square inch overcoming a lower, but by a certain pressure on a large area overcoming the same pressure per square inch, or even a greater one, on a smaller area. This leaves out of consideration the atmospheric pressure on the trunk in those engines which have a trunk. In the other class of engines where this atmospheric pressure is removed, the two pistons are treated as one, which in fact they are, since the intermediate space is not used for steam. It makes no difference whether the effective area of one side of a piston is diminished by a trunk, or by a diminution of the diameter of the cylinder on that one side alone, the latter rendering it necessary to put a distance at least equal to the stroke between the two working sides of the

piston. In these engines the adjacent sides of the pistons might be used instead of the opposite ones, by putting in a division between the cylinders, and by connecting the outer ends with the condenser. The action would then be precisely as before; but there is no reason why the one arrangement of working should *replace* the other; the two might take place simultaneously, and then the engine becomes simply one of that class first described as without a receiver. Where steam is thus used on both sides of each piston the exchange of the back-pressure lines in the diagrams makes very little difference, except that it shows the effective pressure of steam in the small cylinder to be more equable throughout the stroke than would appear from the ordinary diagram; the greatest amount of back pressure coinciding in time with the highest steam pressure, as both occur at the beginning of the stroke. The low-pressure steam line should coincide with the high-pressure exhaust, since there is free communication, and any fall of pressure between the cylinders must be due to throttling in the passages.

It has been remarked that in compound engines provided with a receiver, the work of admission to the large cylinder is sometimes due partly to intermediate expansion, but always partly, and sometimes entirely, to direct transfer of work from the small piston. In the continuous-expansion compounds without a receiver this work of admission, transferred directly from one piston to the other, occurs throughout the low-pressure stroke, simultaneously with the work due to expansion,

and consequently it is not distinguishable from the latter in the diagram.

There is another form of compound engine, if such it may be called, to which the term "*continuous-expansion engine*" has been especially applied. It has two cylinders placed side by side, and the cranks are at right angles with each other. Steam is admitted to the high-pressure cylinder during something less than the half stroke. At this point, or just before it, the low-pressure piston being then at the beginning of its stroke, a communication is opened between the two cylinders through the back of the low-pressure valve and through ports formed in the side of the small cylinder at about half stroke. The steam is now free to expand in both cylinders during the remainder of the high-pressure stroke, at the end of which time the low will have reached its half stroke. Instead, however, of the high-pressure cylinder then opening at once to exhaust, the steam is retained in it for a short time, during which expansion of the steam in both cylinders continues in consequence of the advance of the large piston, which is travelling at this time at its maximum velocity; the small one, on the other hand, being nearly stationary. When, however, the low piston reaches its three-quarter stroke, or thereabouts, the communication between the cylinders is closed by the low valve, and immediately afterwards the high cylinder exhausts into the condenser. Expansion is still continued in the low cylinder until the end of its stroke, when it, too, exhausts into the condenser.

The advantage claimed for engines built upon this

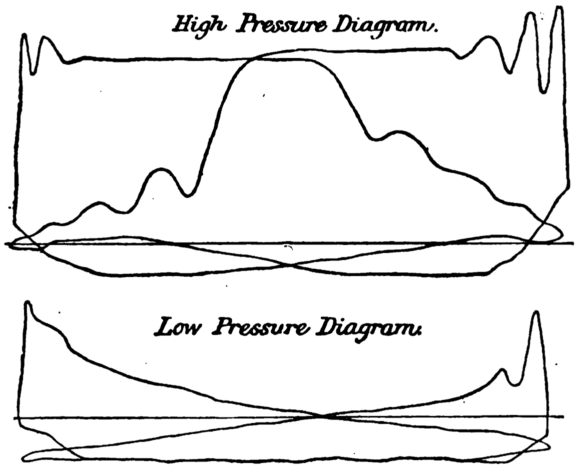
system over non-compounds is that any required rate of expansion may be obtained without the waste of steam which takes place in the passages and clearance of a single cylinder with an early cut-off. Further reference will be made to this waste subsequently.

Again, the advantage over compounds lies in obtaining continuous expansion to any desired extent with cranks at right angles and without the use of extra valves and eccentrics. Three valves only are required, namely, a main valve for each cylinder, and a small valve for retarding the high-pressure exhaust. An expansion valve may, however, be beneficial on the small cylinder. Provision is made in these engines for rendering the cylinders independent at a moment's notice, both cylinders then taking steam direct from the boiler. This is a great convenience in the case, for instance, of vessels coming into port.

The disadvantages of the system appear to be that both cylinders are subjected to considerable variation of temperature and pressure. Both receive steam of pressure nearly equal to that in the boiler, and both ultimately communicate with the condenser, so that the loss of heat by radiation, &c., during the exhaust must be appreciable. The strain also at the time of the opening of communication between the cylinders must be very great, as both pistons are under the pressure of unexpanded steam. It has been found in practice that the horse-power developed from the high-pressure cylinder is sometimes decidedly in excess of that from the low, but this would not be a very serious drawback *in most* cases.

The diagrams taken from the continuous-expansion engines, of which Fig. 14 is a copy, present no peculiarities except the very rapid fall of pressure after the

FIG. 14.



half stroke in the high-pressure cylinder, and from the beginning of the stroke in the low. The repression of the exhaust from the high is also very clearly shown.

The foregoing classification of compound engines according to their mode of expanding the steam, and with special reference to the avoidance of intermediate drop, includes the chief varieties of form now in use.

A large proportion of marine engines belong to the class first noticed, namely, those which have an intermediate receiver, with parallel cylinders and cranks at right angles or nearly so. Many stationary engines are also made on this principle. In Cowper's engine the

receiver is a separate vessel, and reheating takes place in it.

Beam engines generally belong to the class next described, with no receiver and with pistons moving simultaneously. In Woolf's engine, one of the earliest forms of compound, both the cylinders act on the same end of the beam; while in MacNaught's they are placed on opposite sides of the main centre, thus diminishing the strains on the beam, but requiring longer connecting passages for the steam. Annular cylinder engines belong to the same class.

The receiverless engines, with cranks at an angle a little short of 180°, sometimes known as Craddock's, are common in the mercantile navy, but are little used on land.

Perkin's high-pressure compound engine represents the continuous-expansion class, having two pistons on one rod, the steam acting on one side only of each piston.

The engine last described, in which communication is established between the cylinders at the commencement of the low stroke, and at half stroke in the high, will be recognized at once as Stewart's continuous-expansion engine.

CHAPTER VIII.

SIZE OF RECEIVER.

In the previous chapters the intermediate receiver has been supposed to be indefinitely large, because its size did not affect the questions then under examination. It has been seen already that a receiver can be dispensed with in some engines, namely, those in which steam can enter the large cylinder during the whole time of its exhausting from the small one. When the time of demand coincides with that of supply, no warehouse is needed. In engines, however, in which the high-pressure cylinder continues to exhaust at a time when the low cylinder is not open to steam, a receiver of some sort is absolutely necessary; and if no special one is provided, the passages and low-pressure steam chest will perform its functions. In such circumstances, when a receiver is required but is at a minimum, the variation of back pressure in the high and admission in the low cylinder will be excessive; and when the receiver is very large this variation will be insignificant. *This variation is the obvious and the only result of the size of the receiver. It is particularly important to remember that the intermediate expansion, although taking place in the receiver, does not at all depend in amount upon its size.* Given the volume and pressure of the steam which enters the receiver at each stroke,

and the volume which leaves it, the pressure of the latter can be found, as before explained, without reference to any other circumstances of the case. The variation then of the pressure in the receiver,—which variation is inversely proportional to its size, other things being equal,—does not directly affect the economy of the engine, although it impairs the neatness of the diagrams by giving an unsteady exhaust line in the high and steam line in the low pressure card, and it tends therefore to irregularity of strain. A good-sized receiver will therefore be an advantage where it can be obtained easily, but no great loss will result from having only a small one.

The determination of the variation of pressure in the receiver involves a long and intricate calculation, which is rarely undertaken in practice, because the result is of little value when obtained; it requires also to be gone through from the beginning in every separate case, and accuracy must be sacrificed to some extent by neglecting a few of the details of the case if the calculation is to be kept within any reasonable bounds. Some notion, however, of the extent of the variation may be obtained from a simple example, in which it is assumed that the large cylinder cuts off steam exactly at the instant that the high-pressure cylinder opens to exhaust, the small piston being at that time at the end of its stroke. Expansion is considered as taking place by Boyle's law, the product of volume and pressure being supposed constant.

Let h = capacity of high-pressure cylinder,
r = capacity of receiver.

The capacity of the large cylinder will not be required, but its capacity up to the point of cut-off must be known. Let this be called k.

Let p_1 = terminal pressure in high cylinder,
p_2 = pressure in receiver when large cylinder is cutting off steam,
p_3 = " " just after high exhausts,
p_4 = " " just before low takes steam.

Of these p_1 is known, and the rest require to be found. Since the amount, by weight, of steam entering and leaving the receiver at each stroke is the same, therefore

$$p_1 h = p_2 k,$$

$$p_2 = p_1 \frac{h}{k}. \qquad [1]$$

This gives the pressure when the low valve is closing. Immediately afterwards the high cylinder exhausts, this operation being a mixing together of a receiver-full of steam at pressure p_2 and a small cylinder-full at p_1, the whole finally occupying a volume $h + r$. The pressure which results will therefore be

$$p_3 = \frac{p_2 r + p_1 h}{r + h},$$

$$= \frac{p_2 r + p_2 k}{r + h},$$

$$= p_2 \frac{r + k}{r + h}. \qquad [2]$$

The volume of this steam is then diminished by the advance of the small piston, until the low valve opens. Let the portion of the small cylinder which is still in communication with the receiver at the time of the large cylinder taking steam, have a capacity $x h$, then

the volume of the amount of steam in equation [2] will be reduced to $r + x h$, and therefore

$$p_4 = p_2 \frac{r + k}{r + x h}. \qquad [3]$$

This will be the maximum pressure in the receiver in ordinary circumstances, although where the low cylinder is not much larger than the high, the greater velocity of the high piston may produce a slightly greater compression.

In the completion of the high-pressure stroke the alteration of volume which the steam undergoes is from $r + x h$ to $r + k$, since by the time the high stroke is ended admission to the low cylinder becomes complete also; and thus the pressure is reduced to p_2 again, which is the minimum pressure in the receiver. The proportion therefore of the maximum to the minimum pressure in this case is

$$\frac{p_4}{p_2} = \frac{r + k}{r + x h}.$$

As an application of these equations, let $r = 3 h$, $k = 2 h$, and $x h = \frac{1}{2} h$; and let $p_1 = 24$ lbs., then

$$p_2 = 24 \times \frac{h}{2 h} = 12 \text{ lbs.}$$

$$p_3 = 12 \times \frac{3 h + 2 h}{3 h + h} = 15 \text{ lbs.}$$

$$p_4 = 12 \times \frac{3 h + 2 h}{3 h + \frac{1}{2} h} = 17 \text{ lbs.}$$

The variation here is higher than would usually occur in practice, as the conditions of the case all tend to

augment it. If the exhaust from the high-pressure cylinder occurred before the closing of the low valve there would be very little variation. There is one case, however, in which it is sometimes desirable to avoid this, in order to obtain a higher rate of expansion in the low cylinder. It has been stated that the intermediate drop depends solely on the proportion of the low cut-off to the high cylinder. But where wire-drawing occurs by the low valve, the low cut-off must not be regarded as the time when that valve closes, but some earlier time determined by the difference of pressure in the low cylinder and steam chest, since this difference determines the flow of steam through the slowly closing aperture. Consequently if the pressure in the receiver is augmented by the high cylinder exhausting just before the low valve closes, and therefore while wire-drawing is taking place, a practically longer cut-off is obtained in the low cylinder, and consequently there is more loss by intermediate expansion. The loss by wire-drawing also is increased, since it takes place with a greater fall of pressure. In this case therefore it will be advantageous to keep the low cut-off in front of the high exhaust, which may be done by giving the low crank a little extra lead, making the angle of the cranks about 120° or so, instead of 90°. Whether the slight economy effected is worth the sacrifice of smoothness of motion and evenness of strain must depend upon the circumstances of the case.

CHAPTER IX.

EQUALIZATION OF HORSE-POWERS.

IN the ordinary class of compound engines, in which the cylinders drive separate cranks, it is desirable to have nearly equal horse-powers developed from the two cylinders, in order to obtain regularity of motion. This object will be attained by arranging that the mean effective pressures on the pistons shall be inversely proportional to their areas, provided that the stroke of both pistons is the same. The mean effective pressure in each cylinder depends upon the proportion of the rate of expansion done in that cylinder to the total rate, and to that done in the other cylinder. These rates of expansion depend, among other things, on the proportion of the areas of the pistons. The loss by back pressure in the condenser, as well as the loss by partial condensation, or, at any rate, diminution of temperature of the steam during expansion, introduces further complications into the problem of equalization of the horse-powers, and they concur in rendering a general and satisfactory solution of it impossible.

If, however, we neglect the two last-named considerations, and suppose intermediate expansion to be avoided, the case becomes very simple, and an inquiry into it may throw some light on the more intricate cases which occur in practice.

Call R the total ratio of expansion, the product of the ratios in the high (r_1), and in the low (r_2); and call the areas of the high and low pistons a_1 and a_2 respectively; the *mean effective pressure* upon each p_1 and p_2 respectively; and the initial pressure in the small cylinder P; then since there is no intermediate expansion

$$\mathrm{R} = r_1 \times r_2$$

$$\text{and } a_2 = a_1 r_2,$$

$$\text{and } \frac{\mathrm{P}}{r_1} = \text{terminal high} = \text{initial low.}$$

Applying the ordinary formula for mean pressure, we have that in the small cylinder equal to.

$$\frac{\mathrm{P}}{r_1}(1 + \log. r_1).$$

To obtain the mean *effective* pressure, the back pressure, which is equal to the initial low, must be deducted. Then

$$p_1 = \frac{\mathrm{P}}{r_1}(1 + \log. r_1) - \frac{\mathrm{P}}{r_1}.$$

$$= \frac{\mathrm{P}}{r_1}(\log. r_1).$$

Similarly in the low-pressure cylinder

$$p_2 = \frac{\mathrm{P}}{r_1 r_2}(1 + \log. r_2).$$

When equal horse-powers are to be obtained,

$$a_1 p_1 = a_2 p_2,$$

$$= a_1 r_2 p_2;$$

$$\text{therefore } \frac{p_1}{r_2} = p_2.$$

Substituting in this equation the two values just found for p_1 and p_2 we have,

$$\frac{\mathrm{P}}{r_1 r_2}\log. r_1 = \frac{\mathrm{P}}{r_1 r_2}(1 + \log. r_2),$$

$$\log. r_1 = 1 + \log. r_2,$$

$$r_1 = r_2 \times e,$$

where $e = 2{\cdot}7183$, the base of hyperbolic logarithms. And since

$$\mathrm{R} = r_1 \times r_2,$$

therefore

$$r_1^2 = \mathrm{R} \times e \qquad r_1 = \sqrt{\mathrm{R} \times e}$$

$$r_2^2 = \frac{\mathrm{R}}{e} \qquad r_2 = \sqrt{\frac{\mathrm{R}}{e}}.$$

Hence when equal horse-powers are obtained under the conditions given above, the rate of expansion in the large cylinder, which is also the proportion between the cylinders, will be found by dividing the total rate of expansion by 2·7183 *and taking the square root of the result.* For instance, if 10·8 be the total ratio of expansion, the ratè in the low found by this rule will be 2, which will also be the proportion between the areas of the cylinders. And for any total ratio under 10·8 this proportion should be less than 2, a conclusion differing widely from those arrived at by practice. The cause of this disagreement is clear, for all the conditions disregarded in this case, but which occur in practice, tend in one direction. *Back pressure* in the condenser diminishes the effective pressure in, and therefore the work obtained from, the large cylinder, and to an extent proportional to its area. *Condensation* of the steam during expansion has the effect of giving a higher total rate of expansion than that found by dividing the capacity of the large cylinder by that of the high cut-off; and the loss due to cooling accumulates during expansion, and is therefore greater in the low than in the high pressure cylinder. *Intermediate expansion*, where it takes place, adds to the power obtained from the small cylinder, and even if the rate of expansion in each cylinder remains unaltered, the total rate only being increased, the loss due to back pressure in the large cylinder will be slightly increased, as already pointed out.

On all these grounds therefore a much higher rate of expansion in the large cylinder than that given by the

formula is necessary when equal horse-powers are to be obtained. Intermediate expansion serves as a means of adjustment of the horse-powers, and where a cut-off valve, or even a reversing link, is provided for the large cylinder, the rates of expansion in it and in the receiver can be so arranged that nearly equal powers shall be given whatever the proportion of the cylinders. Where, however, the rate of expansion in the low is at most *two*, as will be the case when no cut-off valve or link occurs, either the rate of expansion in the high must also be small, or else the proportion of the cylinders must be kept down, say, to $2\frac{1}{2}$ to 1, in order to avoid much intermediate expansion, as well as a very low terminal pressure in the large cylinder.

The problem of equalization of horse-powers must therefore be worked out separately in each case; and circumstances will determine how far other considerations ought to be sacrificed to this, which will sometimes be of great importance and at other times of hardly any.

The same remark also applies with equal force to a very similar question, namely, that of the equalization of the strains upon the cranks. These will be found by multiplying the area of each cylinder in inches by the *maximum effective* pressure per square inch which occurs in that cylinder. This maximum pressure will generally occur at the beginning of the stroke in the small cylinder, but in some cases it may occur just after, where the exhaust is retarded, as in Stewart's engine, or even as in those known as Craddock's. In finding the strains on the cranks from diagrams, it is essential

to remember to reverse the back-pressure lines, as before mentioned, otherwise cushioning, which occurs before the stroke finishes, appears to deduct from the admission pressure which occurs at quite another time.

In the low-pressure cylinder the initial pressure is often the maximum, but with a small receiver the admission line in the diagram is very irregular, and consequently it is sometimes impossible to find theoretically the maximum pressure in a proposed engine.

CHAPTER X.

PASSAGES AND CLEARANCE.

THE discrepancies which occur between the results arrived at by theory and by practice are due to deficiency or error, or both, in the premises upon which the conclusion is based. Even in the simplest mechanical problems it is exceedingly difficult to make proper allowance for all the conditions of the case, and the simplest way to do so appears to be to take each separately, where that is possible, and so avoid complications.

One of the first things which strikes anyone in analyzing an indicator diagram is, that the terminal pressure is usually very much higher than that given by dividing the initial pressure by the proportion of the whole stroke to that part during which steam is admitted. The reason is that this proportion does not truly represent the ratio of expansion, since the length of the stroke multiplied by the area of the cylinder does not represent the total space finally occupied by the steam; nor does the portion of the stroke during which steam is admitted, multiplied by the area of the cylinder, represent the initial volume. A certain amount of space is occupied by steam before the stroke commences, namely, the *passages* or *ports*, and the *clearance* between the piston and the cylinder end or cover. This steam

undergoes expansion with the rest, and its volume therefore must be taken into consideration both before and after expansion. The total initial volume will therefore be the space passed through by the piston up to the point of cut-off, *plus* the capacity of the steam passage and clearance; and similarly the final volume will be the whole space which the piston passes through in its stroke, *plus* the capacity of the steam passage and clearance. The ratio of expansion is therefore the quotient of the latter of these divided by the former. For example, suppose the capacity of the passage and clearance for one end of the cylinder to be ·08 of that of the cylinder itself, that is to say, of the product of the area multiplied by the stroke; then if the cut-off takes place at quarter stroke, the ratio of expansion is not 1 divided by ·25, or 4, but

$$\frac{1 + \cdot 08}{\cdot 25 + \cdot 08} = \frac{108}{33} = 3 \cdot 3 \text{ nearly.}$$

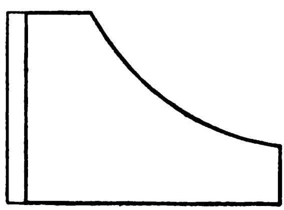
And the earlier the cut-off the more difference does this allowance for passages and clearance make. Thus, if the cut-off is at one-tenth of the stroke in the case given above, the ratio of expansion becomes

$$\frac{1 \cdot 08}{\cdot 18} = 6,$$

instead of 10. *It therefore follows that where a high rate of expansion is required in a single cylinder it is important to decrease the capacity of the steam passages and clearance as far as possible.* And in all cases this space is a source of waste, since at each opening of the exhaust it is

emptied of steam, and requires to be filled up again before the stroke begins. It is true that the steam thus spent acts during the stroke by expansion, but this is only a part of the work which should be obtained from it. While, therefore, the indicator diagram accurately represents by its area the work performed, and by its vertical dimensions the pressure of steam at each point of the stroke, it does not by its horizontal dimensions represent the volume of the steam; and, in order that the latter may be shown, a line must be drawn backwards from the beginning of the stroke, of a length representing the capacity of the passage and clearance, and the diagram will be completed by drawing a rectangle upon this line, as shown in Fig. 15. This modification of the diagram corroborates the remark that the effects of steam in the passage and clearance are more apparent with an early cut-off than with a late one.

FIG. 15.



It must not be concluded because the steam in the passage and clearance of an engine produces more work with a high ratio of expansion than with a low, that therefore the waste due to this cause is less in these circumstances. The work wasted at each stroke, that of admission of the steam in the passages and clearance, is always the same; and as an early cut-off diminishes the total work done per stroke, as well as the steam used, therefore the proportion of the waste to the total is greater with an early than with a late cut-off.

The readiest method of avoiding as far as possible the bad effects of large passages is to shorten them, by using long or double slide-valves, or Corliss gear or the tappet gear applied to beam pumping engines. There is another plan, however, by which the bad effects may be to some extent remedied, in existing engines; and that is by causing the exhaust to close at such a point in the return stroke that the steam so enclosed in the cylinder shall be compressed by the advancing piston, and its pressure thereby raised to nearly that of the boiler by the time that steam is admitted to that end of the cylinder. Fig. 16 shows this action in an ordinary diagram, the area enclosed by the thick line, or diagram

FIG. 16.

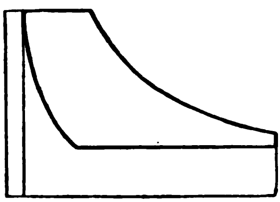
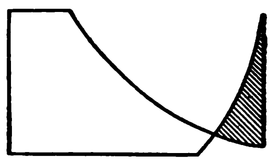


FIG. 17.



proper, being the measure of the work obtained. The precaution, however, of exchanging or reversing the exhaust line must not be neglected; and Fig. 17, which is the result, shows at a glance one great disadvantage of excessive cushioning. The shaded part of this figure represents effective pressure *opposing* the advance of the piston, the back pressure on the exhaust side being in excess of the forward pressure on the other side. The area of this shaded part, therefore, is the measure of the work done by the piston upon the steam, and obtained

from the impetus of the moving parts. It need hardly be remarked that when work is done and undone again there is considerable loss, since friction takes off a percentage in each process. Besides this, great straining of machinery must result from such irregularity of effective pressure as that shown in Fig. 17, which is no misrepresentation of the fact, since it is obvious that the maximum of back pressure does occur at the same time as the minimum of forward, namely, at the end of the stroke. This excessive cushioning is rarely if ever resorted to in practice, and it must be distinguished from the slight cushioning which is almost necessary in a fast-running engine, and the object of which is to overcome to some extent the impetus of the piston and rods, and prevent the sudden shock which would otherwise take place when the valve admits fresh steam suddenly into a nearly empty space against an advancing piston.

The volume of steam contained in the passages and clearance must be taken into account in determining both the ratio of expansion in the cylinder, and also that in the receiver of a compound engine. In the latter case it is evident that the volume of steam liberated by the opening to exhaust of the high-pressure valve, is that contained in the small cylinder and its passage and clearance. When the exhaust closes, however, the passage and clearance remain charged with steam of the pressure of the receiver. Again, the volume of steam abstracted by the low-pressure cylinder at each stroke is that contained in the cylinder up to the

cut-off, and in its passage and clearance. The pressure therefore of the outgoing steam will be to that of the entering in the proportion of the capacity of the small cylinder, *plus* its passage and clearance, to the capacity of the cut-off of the large, *plus* its passage and clearance, *plus* the passage and clearance of the small cylinder. An example will, perhaps, make this more intelligible. If the capacity of the small cylinder, that is to say, the product of its area and its stroke, be called 1, then the volume of the high passage and clearance for one end of the cylinder will be, say, ·07. Suppose the capacity of the large cylinder be 3, and let it cut off at half stroke, so that the capacity of the low cut-off is 1·5; the low passage and clearance may be taken as 6 per cent. of its capacity, which will give ·18; then the proportion of the initial low to the terminal high will be

$$\frac{1 + \cdot 07}{1\cdot 5 + \cdot 18 + \cdot 07} = \frac{1\cdot 07}{1\cdot 75} = \cdot 61,$$

instead of ·66, which it would appear to be if the passages and clearance were not considered. If therefore the terminal pressure in the small cylinder were 20 lbs. the initial pressure in the large would be $20 \times \cdot 61 = 12\cdot 2$ lbs.

It has been remarked that the steam contained in the passages and clearance performs work in expanding, although that which should be done by its admission is wasted. An engine was introduced a few years ago which depended entirely upon the work done by the steam in the clearance of the cylinder; there were no

external passages. The engine was remarkably simple, having neither steam chest, eccentric, nor slide-valve, the duties of all of which were performed by the piston itself. The working barrel of the cylinder had a steam port cast in it on one side, and an eduction port on the other side, both at the half stroke. The piston was a little longer than the stroke, and was deeply recessed at both ends, so that it resembled a piece of pipe plugged in the centre, the plug receiving the piston-rod. Close to each end of the piston were formed in it an eduction port and a steam port, on opposite sides, to correspond to those in the cylinder. The eduction ports each opened through the end of the piston to the *same* end of the cylinder; but the steam ports communicated with passages, cast in the piston, so arranged that the port at the one end of the piston admitted steam to the *other* end of the cylinder. Consequently, when the piston was at the bottom of its stroke the two ports at its top end were opposite to those in the cylinder, and steam was admitted to the under side of the piston, the upper side being open to the exhaust. The up-stroke therefore took place, and at the end of it the conditions were reversed, causing the down-stroke, and so on. It must be observed, however, that the admission of steam occurs in this arrangement as much before one stroke ends as it continues after the next begins. Hence the work done during admission is exactly counterbalanced by that absorbed in overcoming the counter pressure due to lead. But the steam cut off in the cylinder continues to expand during the stroke, and

so gives out a certain amount of available work. This work increases with the quantity of steam used, but a long cut-off which involves an equally early admission is objectionable; consequently the clearance is made excessive, about equal, in fact, to the capacity of the cylinder,—meaning by *capacity* the area multiplied by the stroke, as before. Thus a ratio of expansion of 2 is obtained with an extremely early cut-off, and the mean pressure resulting is about $\frac{5}{8}$ of the initial. An ordinary engine of the same size would be able to keep steam on throughout the stroke, and thus obtain one-fifth more work, with the same expenditure of steam.

While, however, economy in working can hardly be claimed for these engines, they certainly possess many advantages which ought to make them very useful where only a little power is needed. These advantages are economy in prime cost and repairs,—for the engines are not very liable to get out of order;—and the capability of running either way. The engines require to be started by hand, but will run whichever way they are started, because the admission and also the exhaust take place equally on either side of the dead point. In fact, the valve, which in this case is the piston, has a travel equal to the sum of the lead and the lap, or a maximum opening to steam equal to the lead.

This brings us to another case in which similar conditions hold; namely, an engine reversing by means of a rocking link, a locomotive for instance, when the link is put in mid-gear while the engine is in motion. The travel of the valve then becomes the sum of the lap and

the lead, and admission takes place as much before as after the beginning of the stroke. The expansion, however, of the steam in the clearance and passage produces enough effect to keep the engine running in whichever direction it was already moving, if the load upon it is not too heavy. Anyone who has driven such an engine must be familiar with the fact that in order to stop it by means of the link, without entirely shutting off steam, it is necessary to reverse partially for a few moments,—until, in fact, motion ceases in the one direction and is about to commence in the other.

Under these circumstances, the exhaust, as well as the admission, takes place as much before as it continues after the end of the stroke; and therefore when the valve has no *inside* or exhaust lap, the opening and closing will take place at half stroke. This results in cushioning, which in non-condensing engines is very considerable, sometimes raising the pressure in the cylinder before the admission of steam to a point higher than that in the boiler. This is shown in the indicator diagram by a loop at the admission corner, and a loop is sometimes formed also at the exhaust corner, by the expansion of the steam in the cylinder to a pressure below that of the atmosphere. The early opening of the exhaust usually prevents this latter loop, by the admission of air; but where the port opens slowly and not very wide in the end, it fails to supply air rapidly enough to prevent further expansion if the piston is moving quickly.

Whenever an engine is allowed to work expansively

by means of the link, more or less cushioning takes place, together with an early exhaust. These, when not carried to excess, tend to evenness and smoothness of motion; hence the softening of sound which is so noticeable when the link is moved even a trifle out of full gear.

CHAPTER XI.

CONCLUSIONS.

In conclusion it will be well to glance at one or two of the most prominent questions of steam engineering. And, first, as regards the alleged advantages of compound over single cylinder engines, the arguments for the former are so familiar that they need only to be stated. They are, that the economy secured by high pressures and high rates of expansion may be obtained without the drawbacks of (1) excessive strain at the beginning of the stroke, (2) strain of cylinders and loss of effect by exposing the same cylinder to very high and very low pressures and temperatures, (3) loss by passages and clearance, which is unimportant, except with a very early cut-off. Practice alone can ultimately decide the question, and at present opinion sets very strongly in favour of compounds. A certain amount of reaction may, however, be expected before long, and, in fact, the application of the system to very small engines, especially when these are sent to places where skilled workmen are not easily obtained, will tend to bring it about.

The subject of the use of high pressures falls rather under the head of *generation* of steam; for, while its economy is beyond all doubt, its limits are fixed by the

capabilities of safe and trustworthy boilers, and beyond these limits at present (although it may not long continue beyond them) there is another obstacle in the temperature at which lubricants and packing become charred. This may be removed by the disuse of lubricants in the cylinder, and the employment of metallic packing.

The question of the proportions of the cylinders of non-continuous expansion compound engines has been already referred to, and the principles arrived at may be embodied in the following general rule. If a high ratio of expansion is used in the small cylinder, by the employment of an expansion valve, then intermediate expansion should be reduced to a minimum in order to equalize the horse-powers, and also to avoid the slight loss of power due to it. This must be done either by applying an expansion valve to the large cylinder, or else by making the proportion of the areas small, that is about 2¼ or 2½ to 1. When, however, no great amount of expansion takes place in the high-pressure cylinder, a considerable intermediate drop is advantageous, in order to equalize the horse-powers and strains, and also to prevent waste by a too high terminal pressure in the large cylinder. In this case therefore the proportion should be comparatively large, 4 to 1 being a very good ratio. The proportions which find favour in practice are very various, owing probably to such differences of conditions as those just mentioned. Among marine engines, the lowest proportions occur principally on the Thames and the Mersey, 2½ to 1 being not uncommon. In Scotland a

slightly larger proportion of about 3 or $3\frac{1}{2}$ to 1 is perhaps commonest; but here again there is great variety. On the Tyne, however, 4 to 1 prevails with remarkable uniformity, and as much as $5\frac{3}{4}$ to 1 has been tried, if not a higher proportion. The Tees also favours 4 to 1 for the most part.

It may be useful to give an example of the calculation by which the diameters of the cylinders of compound engines are determined. We may assume the circumstances of the case to be as follows:—The indicated horse-power required, including all necessary allowances, is to be 400; the piston speed is 380 feet a minute, and the pressure of steam when it reaches the engines 70 lbs. by the gauge, or 85 lbs. absolute. The steam will be expanded *three* times in the high-pressure cylinder, by means of an expansion valve, which is provided in order that when the pressure on the boilers has to be reduced the same amount of work may be got out of the engines, though not quite so economically, by lengthening the admission. Assuming that it is not important in this case to aim at equality either of horse-powers or strains in the connecting rods, there will be no expansion valve on the low-pressure cylinder, but steam will be cut off at about half stroke by an ordinary valve, double-ported if necessary. For such an engine a good proportion for the cylinders will be 3 to 1, by the adoption of which proportion a total ratio of expansion of 9 will be obtained. From these data and from the tables for mean and terminal pressures of steam expanding without receiving heat, which will be suffi-

ciently accurate even if a steam jacket is used, the following values are found:—

Initial pressure in small cylinder = 85 lbs.
Mean „ „ „ = 57·6 lbs.
Terminal „ „ „ = 25 lbs.

Intermediate expansion takes place in the proportion of 2 to 3, the proportion of the small cylinder to the large up to the point of cut-off. This gives

Initial pressure in large cylinder = 16 lbs.
Mean „ „ „ = 13·3 lbs.
Terminal „ „ „ = 7·4 lbs.

Subtracting from the mean pressures in the high and low cylinders the back pressures, which are 16 lbs. and, say, 2 lbs. respectively, we obtain

Mean effective pressure in small cylinder = 41·6 lbs.
„ „ „ large „ = 11·3 lbs.

For every square inch of area of small cylinder there are three square inches of area of large, and consequently for every square inch of small cylinder there is a mean pressure of $11 \cdot 3 \times 3 = 33 \cdot 9$ lbs. in the low-pressure piston-rod, which, added to that of 41·6 lbs. in the high-pressure rod, gives a total mean pressure of 75·5 lbs. for every square inch of small cylinder.

Turning now to the horse-power required, *viz.* 400, we have

$$400 \times 33{,}000 = 13{,}200{,}000 = \text{foot lbs. per minute required.}$$

Dividing this by the piston speed, which is 380 feet a minute, the total mean pressure is obtained—

$$\frac{13{,}200{,}000}{380} = 34{,}737 \text{ lbs.} = \text{total mean pressure;}$$

and this divided again by 75·5 lbs., the total mean pressure for every square inch of small cylinder, gives the area of small cylinder in square inches—

$$\frac{34{,}737}{75 \cdot 5} = 460 = \text{area of small cylinder};$$

consequently

$$1{,}380 = \text{,, large ,,}$$

which areas correspond respectively to $24\frac{1}{4}$ inches and 42 inches diameter.

The horse-powers obtained from the two cylinders will be in the proportion of the mean effective pressures in the two cylinders multiplied by their proportional areas, or in this case as 41·6 is to 33·9. In a similar manner the proportion of the maximum strains in the connecting rods will be found to be as $85 - 16 = 69$ is to $(16 - 2) \times 3 = 42$. The inequality would be less than this however in practice, owing to the variation of pressure in the receiver. It has been explained that this variation, the extent of which depends upon the size of the receiver, adds to the strain and to the work done in the low-pressure cylinder, while it reduces the amount of both in the high; and consequently if the receiver were not large the inequality of the horse-powers would be very slight. Exact equality may be obtained with a large receiver by diminishing the large cylinder, and so sacrificing the excess of power developed in the small cylinder; or, secondly, by a later cut-off in the small cylinder; or, thirdly, by an earlier cut-off in the low-pressure cylinder, which will involve the addition of an expansion valve.

The consumption of steam in an engine is found by multiplying the area of the cylinder by the piston speed and dividing by the ratio of expansion. In the example just given the cubic feet of steam of boiler pressure consumed per minute is

$$\frac{3 \cdot 19 \times 380}{3} = 404.$$

Allowance must here be made, however, for the capacity of the passages and clearance of the cylinder, the space occupied by the steam at the time when the valve closes being one-third not of the space traversed by the piston in one stroke, but of that space *plus* the passage and clearance. If, therefore, the passage and clearance for one end of the cylinder are together equal in capacity to 7 per cent. of the space traversed by the piston at each stroke, then an addition of 7 per cent. must be made to the amount of steam used as found above, the result being 432 cubic feet. From the amount of steam required per minute the size of the boilers may be found, allowance being of course made for loss, and for steam used by feed donkeys, &c. The size and cooling surface necessary for the condenser, as well as the amount of circulating or injection water required, will be fixed by the amount of steam exhausting at a pressure of 7·4 lbs., the terminal pressure in the large cylinder. All the data needed for these calculations—the weight, temperature, and latent heat of saturated steam at various pressures—are given in engineering pocket-books.

Table of Constants for Finding Mean and Terminal Pressures of Expanding Steam.

Cut-off.	I. At Constant Temperature.		II. Kept Dry at the Temperature of Saturation.		III. Condensing at the Temperature of Saturation.	
	Mean.	Terminal.	Mean.	Terminal.	Mean.	Terminal.
$\frac{1}{2}$	·847	·5	·839	·479	·833	·463
$\frac{1}{3}$	·7	·333	·687	·311	·678	·295
$\frac{1}{4}$	·597	·25	·582	·229	·571	·214
$\frac{1}{5}$	·522	·2	·506	·181	·495	·167
$\frac{1}{6}$	·465	·167	·449	·149	·437	·137
$\frac{1}{7}$	·421	·143	·405	·126	·393	·115
$\frac{1}{8}$	·385	·125	·369	·11	·357	·099
$\frac{1}{9}$	·355	·111	·339	·097	·328	·087
$\frac{1}{10}$	·330	·1	·314	·087	·303	·077
$\frac{1}{11}$	·309	·091	·293	·077	·282	·07
$\frac{1}{12}$	·290	·083	·275	·071	·264	·063
$\frac{1}{13}$	·274	·077	·259	·065	·249	·058
$\frac{1}{14}$	·26	·071	·245	·06	·235	·053
$\frac{1}{15}$	·247	·067	·233	·056	·223	·049
$\frac{1}{16}$	·236	·062	·222	·052	·212	·046
$\frac{7}{8}$	·992	·875	·991	·868	·991	·862
$\frac{3}{4}$	·966	·75	·964	·737	·962	·726
$\frac{5}{8}$	·919	·625	·914	·607	·911	·593
$\frac{3}{8}$	·743	·375	·732	·353	·723	·336

Initial Pressure × Constant = Mean, or Terminal, Pressure.

Example.—Steam of 75 lbs. absolute initial pressure is expanded in a cylinder without a steam jacket; the true cut-off being $\frac{1}{4}$ of the stroke. As condensation in the cylinder is not prevented, Table III. must be used. The constants opposite to $\frac{1}{4}$ cut-off are ·571 and ·214, which multiplied by 75 give 42·8 and 16 lbs. as the mean and terminal pressures respectively.

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